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MOISTURE in steam is usually caused by particles of water being carried over with the steam bubbles during evaporation, or by condensation in the steam pipes and cylinders.

This moisture is inert, doing absolutely no work and in some cases doing harm.

In small quantities its effect is not noticeable upon the reciprocating engine, except that it increases the steam consumption; but with a steam turbine it causes friction and a pitting action upon the blades.

Separators and superheat are employed to overcome the moisture in the steam, and are a material aid toward that end, but to intelligently cope with this difficulty it is necessary to know the quality of the steam entering the engine or turbine.

Here is where the calorimeter is invaluable.

Like the steam-engine indicator, the calorimeter was formerly regarded by the operating engineer as a laboratory instrument or, at the best, a device to be used only by experts in conducting tests upon a plant. But the calorimeter is now generally regarded as a necessary adjunct to the engine room.

Not only is it indispensable in testing engines and boilers, where its omission would entail erroneous results, but it furnishes a convenient means for determining at any time whether the boiler is supplying sufficiently dry steam.

The accuracy of a calorimeter depends upon the manner in which it is attached to the steam pipe. Many an instrument has been unjustly condemned when the fault lay with the man who connected it to the steam pipe.

The quality varies in different portions of a pipe; this variation being greater in a horizontal than in a vertical pipe. Steam drawn from a point near the inside surface is apt to contain more moisture than that taken from the center of the pipe.

One of the best methods of obtaining a fair sample of steam is to cut a long thread on a ½-inch nipple, into which a series of ¼-inch holes have been drilled. Let this nipple project into the steam pipe about three-fourths of the diameter, and as near as possible to the point at which the quality is desired. Care should be taken to place the calorimeter close to the steam pipe to avoid radiation losses, and an ample flow of steam should be allowed through the instrument.

More accurate methods have been devised in the form of sampling tubes, which are more or less complicated, but for all practical purposes the above method will suffice.

There are several types of calorimeters which are divided into classes depending upon their mode of operation. These are the condensing, separating, superheating and electrical calorimeters. Each has its field of application, but the best makes of the three types last mentioned agree very closely as to results, and apparently are much more reliable than one of our contributors will admit.

It may be worth while to "dig up" on this subject and become as familiar with the calorimeter as most engineers are with the indicator.

The Manufacture of Coal Briquets

By A. R. Maujer

Coal dust is dried, ground to uniform size, mixed with a suitable binder, pressed into briquets and baked to remove objectionable odor. The entire operation is continuous and automatic.

The difficulty of burning coal in the form of dust, especially in the cities, where smoke is objectionable, is so great as practically to prohibit its use. Until a comparatively recent date the price of coal in convenient form has been sufficiently low to make it unprofitable to spend any time or effort in making coal dust available for use. As a consequence young mountains of culm have grouped themselves about the mouths of the coal mines to remain there as monuments of man's prodigality of nature's store. As the price of coal advanced, due to the increase in the rate of diminution of the known supply, the increase in the demand and the rise in the costs of production and transportation, the efforts to make coal dust available for use were stimulated in proportion to the advance.

FIG. 1. GENERAL VIEW OF THE PLANT

An obvious way in which to make culm suitable for use, is, of course, to mold it into lumps or briquets of desirable size. This is done by mixing with the culm some suitable material to cement or bind the particles of coal together and pressing the mixture into the desired shape.

The chief difficulty usually encountered was in securing an efficient binder. A suitable binder should be capable of forming briquets which will not break easily in handling, should have no bad effects upon the nature of the ash, should not diminish unreasonably the heat value of the finished product, should not cause smoke or difficulty in combustion and, preferably, should not cause the briquets to have any undesirable odor.

The Economy Coal Company, of New York City, of which brief mention was made in the June 21 issue of POWER, holds the patents issued to Miss Pauline Grayson, the inventor of a briquetting process and a binder of special composition which is said to possess the qualities just enumerated.

The No. 1 plant of the company is located at the foot of West Forty-eighth street on the river front. A view from the river is given in Fig. 1. The wooden bin, a corner of which may be seen to the right, contains the coal dust. To the left of the dust bin a new storage bin for the finished briquets is to be erected. This is to be of concrete and steel construction.

The small addition at the left contains the boiler and engine rooms. The main engine is a 175-horsepower Corliss. The two steel tanks to be seen at the sides of this addition are the receiving tanks for the binder.

At the left of Fig. 2 is shown the grinder, and in the background the drying kiln, extending across the room and along the left wall to the inclosed bucket elevator which may be seen at the far side of the grinder. A screw conveyer in the floor of the dust bin pushes the coal dust onto a short bucket elevator which empties into a hopper located on top of the kiln just beyond the limit of the view at the right. The hopper discharges onto a belt conveyer which carries the culm to the aforementioned elevator which discharges it to the grinder through the 45-degree chute, as shown. The temperature in the kiln is maintained at about 400

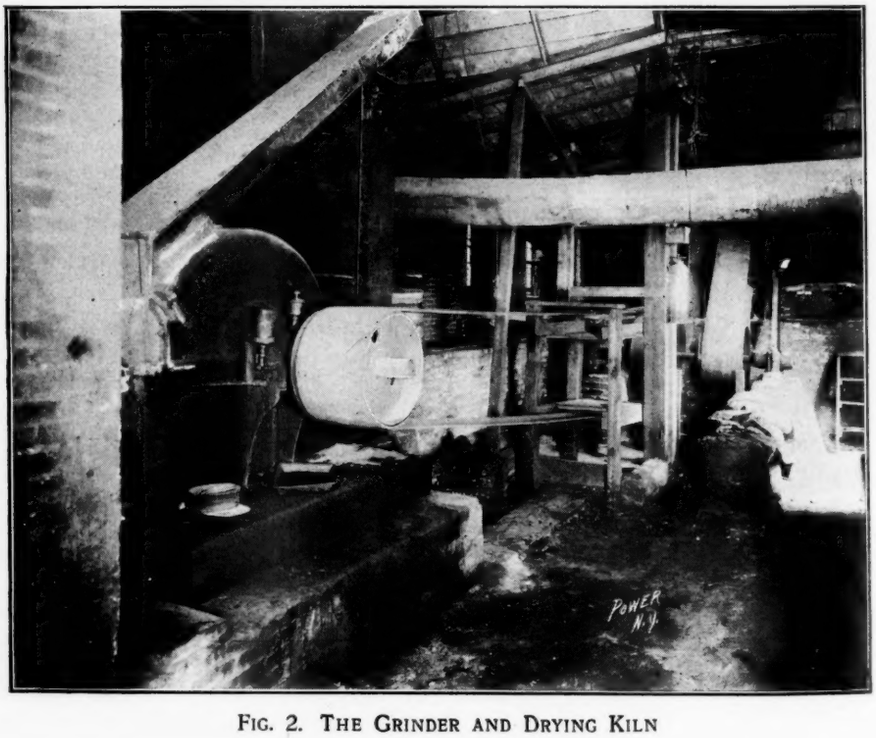


FIG. 2. THE GRINDER AND DRYING KILN

degrees Fahrenheit, so that the excess of moisture in the dust is all driven off by the time it reaches the grinder. In the grinder all of the lumps are reduced so that they will pass through a screen having an ⅛-inch mesh.

tanks to the distributing tanks shown in the same view and in Fig. 4.

The mixing troughs are shown in Fig. 4. The upper one is 10 feet long. The lower one, in which the bulk of the binder is added, is 40 feet long. The

rollers which press it into briquets. Part of one of the rollers may be seen in the middle of the press just beneath the chute.

The briquets fall onto the belt conveyer which passes under the press and

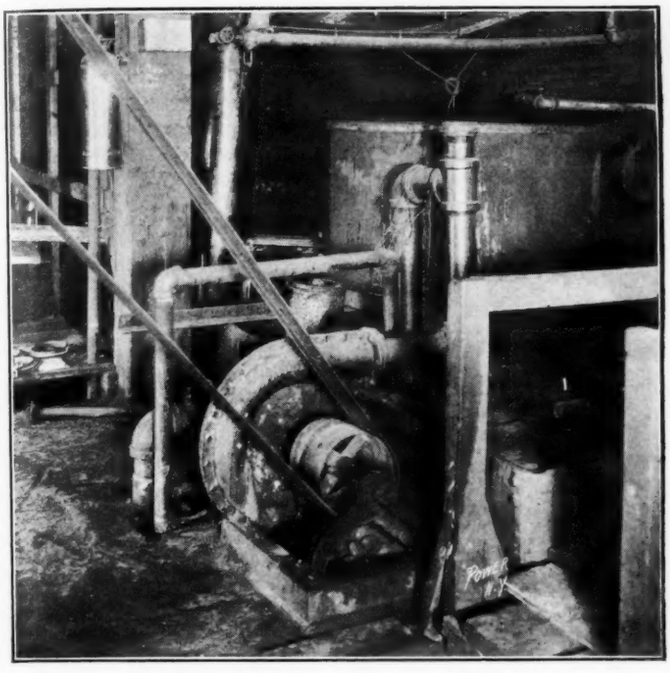


FIG. 3. DISTRIBUTING TANKS AND PUMP

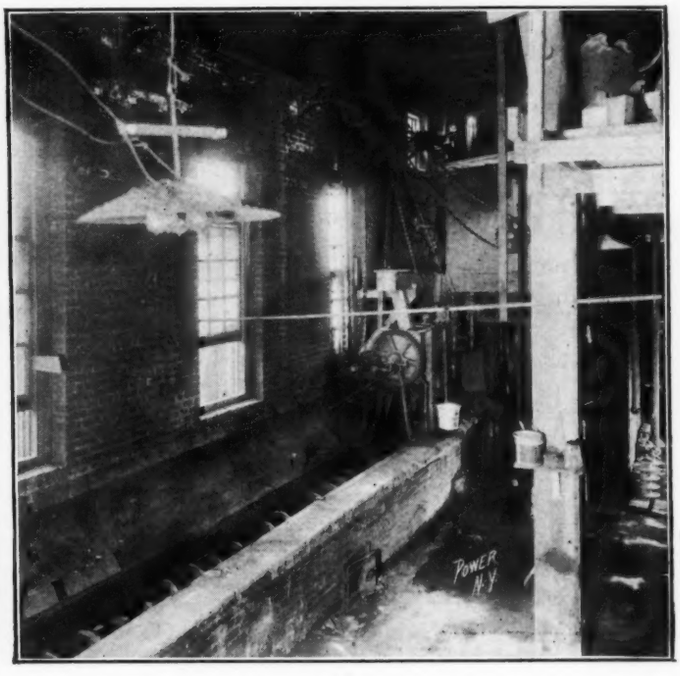


FIG. 4. THE MIXING TROUGHS

A second bucket elevator, which is hidden behind the brick pier at the extreme left of the view, raises the culm to the first of the two mixing troughs in which the binder is added.

cover of this trough has been removed to afford a view of the mixing blades.

This trough empties the mixture of dust and binder into a pit from which it is raised by means of a bucket ele-

which may be seen in Fig. 6. From this conveyer the briquets are raised to another one on the same floor but at a higher level and traveling in the opposite direction. This conveyer empties the

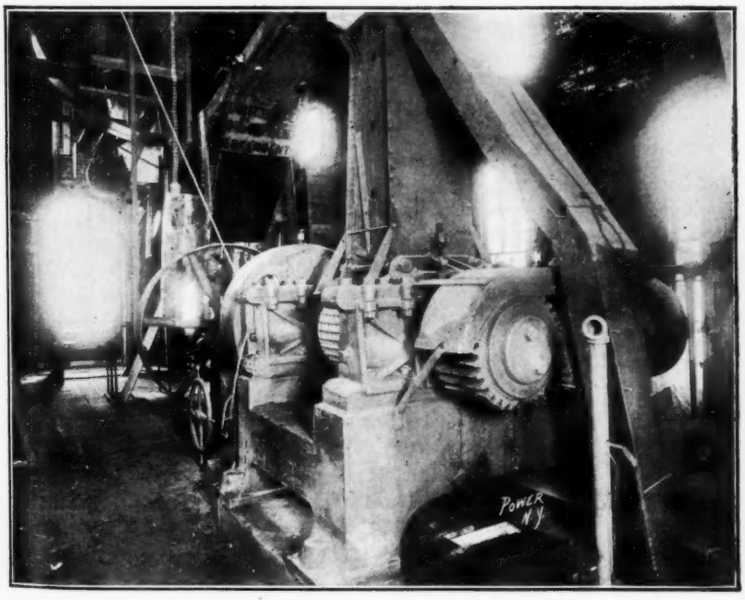


FIG. 5. ONE OF THE PRESSES

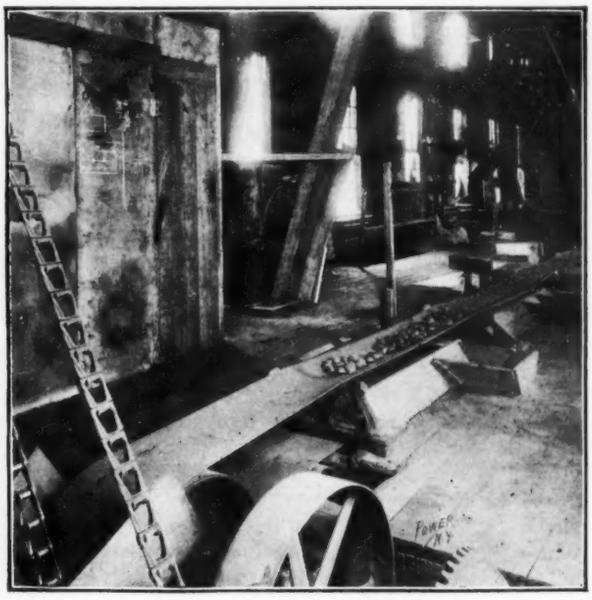


FIG. 6. FINISHED BRIQUETS ON THE CONVEYER

Fig. 3 shows the mezzanine floor which is located just above the upper trough. A portion of this floor may also be seen at the upper right-hand corner of Fig. 4. The centrifugal pump seen in Fig. 3 pumps the binder from the receiving

vator to the floor above, upon which the presses are located.

One of the presses is shown in Fig. 5. The mixture passes along on a belt conveyer and drops down through the sheet-iron chute above the press to the

briquets into a hopper which passes them into the oven in which they are baked for about 15 to 20 minutes. The object of the baking is to deodorize the briquets and drive off the smoke producing volatile matter.

As the briquets leave the hopper they fall onto a wide, slowly moving conveyer made up of lengths of ¼-inch wrought-iron pipe spaced 1¼ inches apart. The oven is located directly over the boilers in the power house. The waste gases from the boiler furnaces are passed through the oven and furnish the heat to do the baking. Falling off the conveyer at the end, the briquets pass through the outlet hopper onto another conveyer which distributes them in the storage bin.

The entire operation is continuous and automatic. Except for getting the culm out of the barges, no hand labor is required in the process. Adjusting devices are located at various points so that the speed of each step can be synchronized with all of the others and the entire system can by this means be kept at its most efficient working speed. The present capacity of this plant is about 12 tons per hour.

Briquets can be made from any kind of coal dust. At present, however, the principal product used is Pennsylvania anthracite dust, of which large quantities are available at a reasonable charge for both coal and freight.

We are indebted to Dr. L. H. Thomas, secretary of the company, for his courtesy in extending to us the priviledge of publishing this description. We acknowledge the assistance given and courtesy shown by Charles A. Hercht, general superintendent of the company.

Water in Steam May Cause Trouble

By W. H. Booth

Arrange the boiler layout with the stop valve near the crown, slope the piping toward the engine and avoid the use of all drains but the one from the separator near the engine.

If a steam pipe has a slope back to the boiler, it is likely to be at a small gradient, only, and the effect of gravity which tends to send back to the boiler any water formed in the pipe is more or less counteracted by pipe roughness. The water flows backward with its upper surface moving faster than its lower part, and the movement is against the direction of flow of the steam. Unless the flow of steam is very slow it is practically impossible for the water to flow down the small descent to the boiler. Some of it is sure to be whisked up by the steam and carried toward the engine, and ultimately there will come a time when a plug of water will be projected forward to the engine, of a size large enough, perhaps, to do mischief. Both experience and reason point to the system of sloping pipes toward the engine, thereby helping the flow of steam to hurry forward all wetness into the separator. So much is this the case that the boiler stop valve should be placed at the highest part of the steam range. This means that the valve must be higher than the engine and that the valve must stand at the head of a pipe branching out vertically from the boiler. Some condensation must occur in this length, but it has the full aid of gravity to return it to the boiler, and if any of it is carried off by the steam it passes through the valve and thence descends to the engine.

When the valve is closed it is free from water. If placed close down to the boiler, with a vertical branch between the valve and the steam main, then when the valve is shut the pipe becomes full of water above the valve. When the valve is opened, the water may be projected violently toward the engine along the empty and perhaps vacuous main. Or, the main may already be in communication with other boilers. In that case the water in the vertical length is probably cool. When the boiler valve is opened, if the boiler is at a lower pressure than the main, the water will be pushed back into the newly opened boiler and all may be well. But it may be that the valve is one which cannot be opened, but must open itself when the pressure below it becomes sufficient to do so. Then when steam does pass the valve it probably lifts the water in the vertical pipe. That portion of the water next the valve is probably hotter than that higher up the pipe. Lifting proceeds without disturbance for a time. Then the column of water breaks; the water mixes with the steam; there is a sudden condensation below the water and a vacuum is formed. The water is driven violently downward by the pressure above and probably the valve is broken. If the valve is opened suddenly and widely this may not occur, for steam flows from the boiler below as quickly as from the main beyond. But if the valve is just "cracked," then a vacuum can form and trouble will ensue.

Kinetic Theory of Gases

As regards the flow of steam and its velocity under such conditions, the kinetic theory of gases is a help in understanding what may happen. According to the kinetic theory, a gas consists of an infinity of particles endowed by heat with the power to move continuously at a high velocity of several hundred feet per second. Obeying the law of motion, they move in straight lines but are in a continuous state of collision with each other and the bounding surfaces around them. But, if by condensation, all particles along a certain section are made suddenly to disappear, a vacuum has been formed and this is simply an absence of live particles. The particles beyond the limit of destruction, having now no opposition, rush into the empty space and drive forward any interfering plugs of water and so produce a destructive water hammer. If the run is long enough, such a plug of water will get up a velocity equal to that of the steam, or several hundred feet per second. No plug of inert water can travel faster than the molecular velocity of the steam, for the steam has a velocity limited by its own physical conditions. But this velocity is so great that immense pressures are created by projected plugs of water.

Where valves are thus set at the base of vertical lengths of pipe, it is proper to apply a drain pipe and a trap. And these drain pipes may fail; they may be frozen, or choked by scale or dirt, and then comes trouble. Clearly, then, the proper thing to do is to arrange everything so as to obviate all drains but the one near the engine, and this should be of fair diameter unlikely to choke. The boiler layout will then properly be such as to give a flow for water toward the engine, while keeping the boiler stop valve near the boiler crown, but not too near. If the steam main is near the stop valve there will be lack of elasticity and good practice will place the valve a couple of feet or so above the boiler crown, with a few feet of pipe connection to the main. Further elasticity may be given by an expansion bend, or U, in the course of this branch, the bend standing out horizontally from the branch pipe and not vertically. Water is then free to flow with no obstruction.

Nowadays, when it is becoming more and more the practice to pay attention to smoke or its nonformation, the raising of the boiler so high above the firing floor is not regarded so seriously. I am speaking now of water-tube boilers which cannot be worked without smoke, unless provided with room sufficient to build in arches and return arches between the tubes and grate. But, placed thus high, the steam-gage and the water-gage glasses are out of proper observation from the floor, and a satisfactory design should insure a platform at a suitable hight from which to attend to such matters.

With shell boilers of small diameter, this question of an upper platform will not arise. The question of water in the steam pipe only exists in the absence of superheat. But even with superheated steam the layout should be as though the steam might be wet.

Few engines will not bear a temperature of 500 degrees Fahrenheit. Even if they will not bear this they will bear some superheat. And if there is any super-

heat at all at the engine, then it is known that the steam is dry. A calorimeter to test steam returns probably does record with approximate accuracy the wetness or the dryness coefficient of the steam that passes through it; but the man is still missing who can devise an instrument or a method by which the calorimeter can be supplied with steam of the same degree of wetness as that which is traveling along the steam main. All calorimeter tests are therefore a delusion, unless they are made on superheated steam, and with superheated steam a thermometer tells all that a calorimeter can tell and tells it with more accuracy. All boiler tests which are calculated out on the basis of a steam production so many per cent. wet, as shown by calorimeter tests, are worthless.

The only method of determining the amount of water carried out of a boiler is to catch it as it drains out of the separator when the boiler is being worked at test rates and pressure, and the steam is getting away at such a rate that the pressure is kept steady. Probably a calorimeter test would show the quality of this steam if taken just past the separator, and the wetness thus found would then be increased by whatever the separator drains showed to be separated from each thousand pounds of steam. The result would not be accurate, but it would be far more approximately true than an ordinary calorimeter test, especially of the kind encountered not long ago, where the testing engineer removed the calorimeter from the branch provided for it and put it in a better place because at the first place the steam came so very wet.

But how did he know that this wet steam was not what the boiler was giving all along? Probably it was so, for the test results were beautifully abnormal and good.

An Example of Lubrication Economy

In a plant equipped with a well-designed oiling system an expert was able to effect a reduction of 42 per cent. in the annual lubrication cost. This was accomplished by doing away with small wastes and a careful selection of the proper oils to meet the various requirements.

With a plant in which the engines, shafting, etc., are well equipped with oiling systems, and where those in charge operate with an economical use of oil, it would seem that the lubrication account must be down to the point of lowest practicable cost. Yet the engineering chemist is able to step in and reduce the lubrication cost by 35 to 50 per cent., without any sacrifice of efficiency.

Two things are probably true in the majority of plants whose owners have attempted to enforce practicable economy: first, real economy in use does not make up for lack of economy in buying lubricants; second, the lubrication expert, because of his more varied experience, can usually discover some substantial economy in usage that has not occurred to the engineers and superintendents, whose intention to economize has not compensated for lack of the technical knowledge that enables the expert to economize.

A striking example of gain in lubrication economy, due to expert technical knowledge of the subject is provided in a report by Arthur D. Little, chemist and engineer, of Boston. This case is the more striking because the plant was very well equipped, and was well managed up to the limit of the technical knowledge of those in charge. During the year preceding this investigation and report, the plant had used 18,500 gallons of various oils, ranging in price from 14 to 49 cents per gallon; and 4675 pounds of grease and gear coating, at prices ranging from 5 to 9½ cents a pound; the total lubrication cost being $4425.

An indefinite amount of waste was due to the absence of a properly equipped central storage place for the oil. Barrels of oil were distributed in different places, and a number of these barrels were found to be leaking. The first remedial step was the construction of an oil house containing metal tanks provided with key faucets, into which the oil could be emptied as soon as received. Provision was made for gaging the barrels as received, and measuring the deficiency of oil in each barrel. This often leads to a substantial saving, but it is rare to find a plant in which it is done. The next step was the introduction of a better system of recording the amounts distributed and the place of use. Previously, upon requisition, oil had been issued by the barrel, but there had been no system of records from which could be made a quick computation of consumption and cost. The new system made it possible to hold each engineer and fireman responsible for his own lubricating costs, and this, of course, tended toward economy. The four large engines in the plant lost about 150 gallons of engine oil per month. The report laid the greater part of this loss to wiping up engine frames and floors with waste, which was then thrown away or burned. To check this loss an oil- and waste-saving machine was recommended, which makes waste fit for repeated use and saves a large proportion of the oil contained in the waste, often as much as 90 per cent.

Economy in purchasing lubricants is reached by following two rules: First, it should be determined just what oils and greases are needed, and the exact chemical and physical properties of each. Second, these necessary qualities should be expressed in specifications, and all lubricants purchased by these specifications, on annual contracts awarded after competitive bidding.

Analysis of the oils and greases used at this plant showed that all were of good quality. Their chemical and physical properties were the same as those which some other plants had succeeded in getting, by buying on specification, at a much lower cost. A cylinder oil, equal in every way to that for which 48 cents had been paid, was obtainable for 25 cents, or even less. The machine oil, costing 33 cents a gallon, and used on the engines, proved to be substantially the same in chemical and physical qualities as the 14½-cent shafting oil. As a substitute for these two oils the report gave specifications for an oil that was obtainable on annual contracts at about the price which had been paid for the shafting oil. It was shown that greases of the desired quality could be bought at from 3 to 5 cents a pound to displace those for which 9½ cents had been paid. Further economy was attained by substituting for the engine and machinery oil used on the slow-speed machinery, a good black oil at about 9 cents a gallon. Similarly, by using a more viscid gear coating during the summer, it was possible to make a reduction of 25 per cent. in the annual consumption.

By reducing the cost of the different oils and greases, by using lower grade oils where these grades were good enough, and by removing certain sources of actual waste, the experts were able to point out a reduction in the annual lubrication cost from $4425 to $2570, or a saving of nearly 42 per cent.

The most important lesson in the facts of this case is that maximum economy in both the use and the purchase of lubricants requires expert chemical knowledge of oils and greases plus a knowledge of the operation and requirements of machinery, and finally, a knowledge of how to buy the required lubricants at the lowest price. The practical value of the lubrication expert depends upon all these things, but in the last analysis his chemical knowledge is the one essential thing that the busy practical engineer or superintendent does not have, and has no time to acquire. Most operating men, having found an oil that serves them well, go on buying that oil because they have nothing but the brand name, or the reputation of the dealer, by which to identify it. When chemistry steps in, the plant ceases to pay for reputation, the lubrication outlay is cut down, and the best oil for the requirements is secured.

Calculating Heat Value of Coal

By Julian C. Smallwood

Calculation of the amount of air required to burn a pound of coal based on either proximate or ultimate analysis of the coal. How to determine the heat value of the coal; the loss of heat through various causes and the "heat balance."

In a previous article a method was given of making calculations from flue-gas analyses based upon the heat value of the carbon burned in the furnace. As stated there, a pound of pure carbon, burned completely to carbon dioxide, yields at the greatest, 14,650 B.t.u. Instead of all this heat being transferred to the water in the boiler, part of it escapes up the chimney as heat added to the flue gas, part of it is lost through incomplete combustion so that not all of the heat that can be is generated, and part of the carbon is not burned at all on account of slipping through the grate and into the ash pit. A list of the various directions in which the available heat of the coal is dispersed is as follows:

Useful heat added to the water, from 50 to 70 per cent. Radiation from boiler and setting, from 5 to 10 per cent. Ashes and unburned carbon in ashes, 5 to 10 per cent. Flue gases and unburned gas in flue, 20 to 40 per cent.

It is with the last item that the flue-gas analysis enables us to deal and, by our calculations, to reduce to the least possible amount. This loss may be further subdivided as follows:

Heat carried up the chimney by the flue gases analyzed and by the moisture in the coal. Loss from incomplete combustion of carbon monoxide and other gases.

The previous calculations have taken no account of the moisture in the flue gas or of combustible material other than carbon which may be in the coal. If something is known about the make-up of the fuel we may include these items in our considerations. Further, if we know something about the actual heat which one pound of the coal will deliver when completely burned (that is, its heat value), we can base our calculations upon this instead of the heat value of carbon. Finally, if we know a little more about the flue gas constituents we shall be able to determine, and therefore regulate, some material losses that may occur. With this in view the calculations to be presented in this article will include such as may be obtained from a usual gas analysis as before described in conjunction with a coal analysis and further calculations from a more complete gas analysis. The methods of obtaining the additional data will be briefly described.

To understand as clearly as possible the process of coal combustion we must first consider in a more detailed way the constituents of the coal. Bituminous, semi-bituminous, and anthracite coals may be distinguished by the amount of so called volatile matter which they contain. This is burnable material which may be driven from the coal by a low heat without causing it to ignite just as sap is driven from some woods in the form of smoke when they are heated to a point just below the burning point. The amount of volatile matter contained in the different kinds of coal when dry is roughly given in Table 1, which also shows the amount of carbon and ash left when this matter is driven off.

TABLE 1. CONSTITUENTS OF DRY COAL.
Per Cent. by Weight.

	Bituminous.	Semi-Bituminous.	Anthracite.
	From	From	From
Volatile matter....	20 to 35	10 to 20	3 to 10
Average.....	30	17	8
Fixed carbon.....	60	74	85
Ash.....	10	9	7

As the volatile matter forms an important part of the heat value of the fuel (particularly of bituminous coals) we should be much interested in burning all of it.

A more complete analysis of coal shows it to consist of the elements shown in Table 2.

TABLE 2. ELEMENTS IN COAL.
Per Cent. by Weight.

Element.	Bituminous.	Semi-Bituminous.	Anthracite.
Carbon (C).....	76	82	89
Hydrogen (H_2).....	5	4	2
Nitrogen (N).....	1.5	1	0.8
Oxygen (O_2).....	7.5	4	1.2
Ash.....	10	9	7

The volatile matter contained in coal is made up of carbon and hydrogen in some form and if it is completely burned the products of combustion are carbon plus oxygen or CO_2 and hydrogen plus oxygen or H_2O. For the last named product two atoms of hydrogen combine with one of oxygen and H_2O or water is formed.

We do not know just how these elements are held by the coal; whether the oxygen and hydrogen are there in the form of oxygen and hydrogen, or whether the hydrogen exists as a chemical compound with part of the carbon. What we do know is that when using bituminous coal, a large amount of hydrogen which, when *freed* from the coal is combined with carbon, is driven off and if this hydrocarbon, as it is called, is not burned under the boiler a considerable loss of heat results. It is therefore an important matter to learn whether there is any hydrogen in our flue gases, particularly when using soft coals.

In Table 1 is given an illustration of the results obtained from what is called the proximate analysis of coal. This can be made quite readily by one not experienced in chemical measurements. As a rule, the determination of moisture is made as well, for all coals contain a certain amount of water, though they may appear to be dry. It is aside from the purpose of this article to tell in detail how coal analyses are made, but the following brief outline of the proximate analysis may be of help. The coal is first sampled carefully by "quartering," a method of getting a small representative sample from a large amount, and then ground, sifted and weighed. The sample is then heated in a platinum pan at a low heat to drive off the moisture. The difference between the original weight and the weight after this heating gives the amount of moisture. The sample is then again heated at a somewhat higher temperature until the hydrocarbons are all driven off, and their amount is determined by the difference in weight. Next, the sample is heated to the burning point and the carbon consumed. The fixed carbon is thus ascertained; the residue is ash, which contains shale, dirt, etc.

In Table 2 are given results of what is called an ultimate analysis. Such an analysis is quite impracticable for any but an experienced chemist to make successfully. It is useful, however, occasionally to have such an analysis made of samples of the coal with a view to checking closely the boiler's performance. The results of such a test generally do not include moisture, the sample being completely dried before testing.

To determine the heat value of the fuel one of two methods may be used. A special test may be made of a sample or the result may be calculated from the proximate or ultimate analyses. The former is the more accurate way and is not difficult, but the apparatus is expensive. The measurement is made by actually burning a small sample of the coal in an air-tight chamber immersed in water. The weight of the water and its rise in temperature due to the combustion being known, the heat generated may be calculated. There are various types of these heat-measuring devices (known as coal calorimeters) on the market and every steam-power plant of any size should have one, if for no other reason than to ascertain definitely the value received for the money put into coal.

To calculate the heat value of a fuel from its chemical analysis, various formulas have been proposed. These are based partly on the fact that the heat

values of the elements composing the coal are known. For example, the heat value of one pound of hydrogen is 62,100 B.t.u. and if we had a coal containing three-quarters carbon and one-quarter hydrogen, we should expect the mixture to yield

$$14{,}650 \times \tfrac{3}{4} + 62{,}100 \times \tfrac{1}{4} = 26{,}500 \text{ B.t.u.}$$

and this would be the case if the carbon and hydrogen existed as such in the coal. This, however, is not so; they are combined as hydrocarbons and otherwise. Now when combustion takes place some heat must be given up to these gases to separate the carbon from the hydrogen so that they may recombine with oxygen. That is, when a chemical compound is broken up into its elements, heat must be added to it; when these original elements combine again with themselves or some other element heat is generated. So there is an interchange of heat in burning coal which makes its heat value dependent upon the previous combination of its elements. We therefore cannot calculate the heat value precisely but certain experimental formulas enable us to make very acceptable estimates of it. One* of them is:

Heat value equals 14,650 times the weight of carbon in one pound of fuel plus 62,100 times the weight of hydrogen in one pound of fuel minus 5400 times the sum of the weights of oxygen and nitrogen.

Taking as an example the data of the complete analysis given later in this article, there results

$$14{,}650 \times 0.837 + 62{,}100 \times 0.047 - 5400 \times 0.058 = 14{,}870 \text{ B.t.u.}$$

Another formula,† for use with the proximate analysis, is:

Heat value equals 146.5 times the per cent. of fixed carbon in one pound of fuel plus K times the per cent. of volatile matter in one pound of fuel.

The quantity K has different values depending upon the amount of volatile matter in the coal and should be selected from the following table:

Volatile Matter. Per Cent.	K.
2 to 15	234
15 to 30	180
30 to 35	171
35 to 40	162

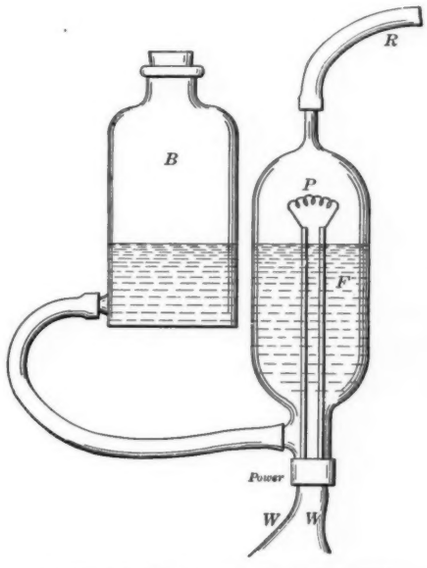
Taking for an example the data from the same sample as was considered in the calculation just made, the heat value is

$$146.5 \times 72 + 180 \times 22 = 14{,}500 \text{ B.t.u.}$$

To return to the flue gases, the moisture they contain can be calculated from the hydrogen and moisture in the coal. The moisture may also be measured directly by the use of special apparatus but this is hardly necessary. The unburned hydrogen in the flue gases may readily be ascertained with a small piece of apparatus to be used with the Orsat in a way which will be described. The only constituents we do not consider, then, are a gas formed by sulphur in the coal whose effect is negligible and, perhaps, some soot or solid carbon due to the too sudden cooling of the glowing particles against a comparatively cool part of the boiler shell.

Briefly, the method of determining the hydrogen is, first to convey the sample, to which some air has been added, to a flask in which all of the hydrocarbons present are completely burned. From the consequent reduction in volume the hydrogen can be calculated. In more detail the process is as follows:

The gas sample is first analyzed for CO_2, CO and O, in the usual manner with the Orsat apparatus. Part of the residue, consisting of nitrogen and hydrocarbons, is then forced out of the measuring tube into the air, 50 cubic centimeters being left. Then, 50 cubic centimeters of air is drawn into the measuring flask and mixed with the 50 cubic centimeters of the sample. The combustion flask F, shown in the figure herewith, is then connected by means of the rubber tube R to the entrance nozzle of the Orsat apparatus. The combustion flask and tubing have previously been filled with water by elevating the bottle B. Having made the connection, the cock between the Orsat apparatus and the combustion flask is opened and by manipulating the two water bottles the sample is slowly forced



COMBUSTION FLASK FOR HYDROGEN DETERMINATION

into the combustion flask. This contains a coil of platinum wire P, the ends W of which are connected to dry batteries or to a properly arranged lighting circuit so as to heat the coil to a bright red heat. As the gas enters the chamber it passes this hot coil and the hydrocarbons are burned to H_2O and CO_2. The residue is then returned to the measuring flask and the diminution in volume is noted, care being taken to allow time for the gases to cool.

Each atom of carbon present before combustion has been taken up by one molecule of oxygen to form CO_2 and there is no reduction In volume due to the burning of the carbon as one molecule of oxygen occupies the same volume as one of CO_2. The hydrogen, however, has formed water which condensed at once so that the total volume before combustion is diminished by the oxygen turned into water. In this combination there is a total of three parts of gas; two of hydrogen and one of oxygen. The hydrogen originally present is, therefore, two-thirds of the reduction in volume. To obtain the amount of hydrogen in the 50 cubic centimeters of the residue, the diminution of volume after treatment in the combustion flask is multiplied by two-thirds. We can then allow for the hydrogen contained in that part of the discarded residue as in the following example:

Fifty cubic centimeters of air added to 50 cubic centimeters of residue. Diminution noted, 0.9 cubic centimeter. Hydrogen in 50 cubic centimeters,

$$\tfrac{2}{3} \times 0.9 = 0.6$$

cubic centimeter. Hydrogen in 81.5 cubic centimeters,

$$\frac{81.5}{50} \times 0.6 = 1$$

cubic centimeter, or 1 per cent.

In testing for hydrogen in this way nothing is learned about the forms in which it appears as hydrocarbons. There are methods by which the free hydrogen and the hydrocarbons can be measured separately, but as the hydrogen alone gives a very good idea of losses due to unburned hydrocarbons, these other more complicated methods are not recommended.

In the process described if there is any ethylene gas (one of the hydrocarbons) present it will be absorbed in the cuprous chloride solution and appear as CO. On this account the determination may not be absolutely exact, but it is close enough for all practical purposes.

Coming now to the calculations to be obtained from the tests, let us assume the following results of coal analyses:

PROXIMATE ANALYSIS.

	Per Cent.
Moisture	1
Volatile matter	22
Fixed carbon	72
Ash	5

ULTIMATE ANALYSIS.

	Per Cent.
Carbon	83.7
Hydrogen	4.7
Nitrogen	1.6
Oxygen	4.2
Ash and sulphur	5.8

and of a flue-gas analysis, $CO_2 = 9$ per cent., $O_2 = 8$ per cent., $CO = 1.5$ per cent., and $N = 81.5$ per cent. Heat value as determined by the calorimeter, 14,800 B.t.u.

AIR THEORETICALLY REQUIRED

Theoretically, only enough air is needed to burn all of the carbon to CO_2

*Mahler's.
†Goutal's.

and all of the hydrogen to H_2O. In one molecule of H_2O there are two atoms of hydrogen weighing one atomic weight each and one atom of oxygen weighing sixteen. Each atom of hydrogen then combines with eight times its weight of oxygen; one pound of hydrogen would require eight pounds of oxygen for complete combustion and so on. To burn the 0.047 of a pound shown by our analysis we must furnish,

$$0.047 \times 8 = 0.376$$

pound of oxygen. To burn the 0.837 of a pound of carbon we need

$$0.837 \times 2\tfrac{2}{3} = 2.232$$

pounds of oxygen since carbon requires $2\frac{2}{3}$ times its weight of oxygen. This makes a total of 2.608 pounds. As the coal contains 0.042 pound of oxygen, only

$$2.608 - 0.042 = 2.566$$

pounds need to be furnished by the air supplied to the boiler. As air weighs four and one-third times the oxygen it contains, the air actually needed is,

$$2.566 \times 4\tfrac{1}{3} = 11.1$$

pounds of air per pound of coal. This calculation is useful to determine how much more air than necessary we are using.

Air Actually Supplied

Calculation to determine the weight of air actually supplied is made, as described in the second article of this series, by first finding the weight supplied per pound of carbon. From the flue-gas analysis, the total atomic weight of oxygen is

$$9 \times 32 + 8 \times 32 + 1.5 \times 16 = 568$$

and the total atomic weight of carbon is

$$9 \times 12 + 1.5 \times 12 = 126$$

and

$$\frac{568}{126} = 4.5$$

pounds of oxygen per pound of carbon, or

$$4\tfrac{1}{3} \times 4.5 = 19.4$$

pounds of air per pound of carbon.

Since there is only 0.837 pound of carbon in a pound of coal,

$$0.837 \times 19.4 = 16.2$$

pounds of air per pound of coal.

So far this calculation does not account for that part of the air used which has been turned into water nor does it account for that part of the carbon which may exist in the flue gases as hydrocarbons. As far as the former is concerned we may assume that all of the hydrogen is changed to water before reaching the stack. In the calculation for the necessary air we find that 0.376 pound of oxygen is needed for the hydrogen. The coal furnished 0.042 pound of this, leaving

$$0.376 - 0.042 = 0.334$$

pound to be furnished by the air. The total air is, therefore,

$$16.2 + 0.334 \times 4\tfrac{1}{3} = 17.7$$

pounds of air per pound of coal.

Loss from Excess Air

The theoretical amount of air necessary was found to be 11.1 pounds. The excess is

$$17.7 - 11.1 = 6.6$$

pounds. If the air is heated from 60 to 560 degrees, the heat added, since the specific heat of air is 0.238, equals

$$0.238 \times 500 \times 6.6 = 785 \text{ B.t.u.}$$

Dividing this by the heat value of the coal gives

$$\frac{785}{14{,}800} = 5.5$$

per cent. loss.

Total Heat up the Stack

The total weight of flue gas that comes from one pound of coal is equal to the sum of the weights of the air fed to the coal and that part of the coal which is burned. That is, the air and the combustible part of the coal which appear as flue gas must retain the weight they had before combining. The analysis of the coal shows it to contain 5 per cent. of ash. If all but the ash is burned, 0.95 pound of each pound of coal passes up the chimney. This added to the weight of air per pound of coal equals the total weight of flue gas to each pound, or

$$0.95 + 17.7 = 18.65$$

pounds of flue gas per pound of coal.

This includes the moisture from chemical union. As 0.047 pound of hydrogen combines with 0.376 pound of oxygen, the moisture resulting is

$$0.047 + 0.376 = 0.42$$

pound. The weight of the flue gas without the moisture is, therefore,

$$18.65 - 0.42 = 18.23$$

pounds. Taking the specific heat of flue gas as 0.235, which is very closely accurate, and the same temperature range as before, the heat added to the gas is

$$0.235 \times 500 \times 18.23 = 2130 \text{ B.t.u.},$$

or

$$\frac{2130}{14{,}800} = 0.142 \text{ or } 14.2$$

per cent. loss.

This calculation is approximate. In order to make an exact determination of this loss it would be necessary to use the specific heat of each of the flue gases and multiply the weight of each constituent by the corresponding specific heat and the temperature range; the sum of these would be the total loss. This, however, is hardly worth while in view of the uncertainty of some of our quantities.

Loss Due to Moisture

The specific heat of moisture, or superheated steam as it becomes, is comparatively high, about 0.48, so that its influence is to increase the heat lost up the stack. Moisture in the flue gases comes from two sources, from the chemical union of the hydrogen in the coal with oxygen and from moisture in the coal which exists in the form of water, as shown by the proximate analysis. Considering the former we do not know just when the chemical combination takes place or the temperature range through which the resulting steam is heated, so it is impracticable to make an accurate calculation for this loss. An easy way of estimating it is to assume that this moisture has heat added to it while in the form of steam through the whole range of temperature. The heat added then is

$$0.48 \times 500 \times 0.42 = 101 \text{ B.t.u.}$$

Considering now the loss due to the moisture shown by the proximate analysis, the moisture is first heated as water from the temperature of the boiler room to the boiling point, then turned into steam, and finally superheated. The specific heat of water is one, and when it reaches the temperature at which it is about to turn into steam at atmospheric pressure, that is, 212 degrees, each pound of it requires 970 B.t.u. to change its molecules from the solid to the gaseous form. The heat added to the 0.01 pound of water in our pound of coal is, therefore, found as follows:

$$\begin{aligned} 1 \times (212 - 60) \times 0.01 &= 1.5 \text{ B.t.u.} \\ 970 \times 0.01 &= 9.7 \text{ B.t.u.} \\ 0.48 \times (500 - 212) \times 0.01 &= 1.4 \text{ B.t.u.} \\ \text{Total} &= 12.6 \text{ B.t.u.} \end{aligned}$$

This added to the 101 B.t.u. previously calculated gives a total of 113 B.t.u. and the loss is

$$\frac{113}{14{,}800} = 0.0076$$

or nearly 0.8 of 1 per cent.

Loss Due to Incomplete Combustion of Carbon

The loss due to the incomplete combustion of carbon is obtained as described in the article preceding this one, except that the heating value of the coal is used. The total atomic weight of the carbon in the CO shown by our analysis is

$$1.5 \times 12 = 18$$

and the heat lost by each pound of carbon is

$$\frac{18}{126} \times (14{,}650 - 4400) = 1460$$

B.t.u. Since there is only 0.837 pound of carbon in each pound of the fuel, the heat lost per pound of coal is

$$\frac{1{,}460}{14{,}800} \times 0.837 = 0.083 \text{ or } 8.3$$

per cent.

None of the foregoing calculations from the flue-gas analysis must be regarded as absolutely accurate because of the assumptions which it is necessary to make. Chief among these is that all of the hydrocarbons are burned to H_2O and CO_2.

We are basing our calculations upon the carbon appearing in the flue gas, but there may be some unaccounted for as hydrocarbons. There is, however, only a small part of the original pound of coal existing in this form, if any, as most of the volatile matter is actually burned completely. All of our calculations as described are sufficiently accurate even if we have the results of a test for hydrogen. If we have such results we may determine the additional loss, previously unnoted, due to incomplete combustion of the hydrocarbons.

Let us suppose that this more complete test has been made and that the following data have been obtained from the same sample as that which we have been considering:

CO_2, 9 per cent.; O_2, 8 per cent.; CO, 1.5 per cent.; N, 80.5 per cent.; H_2, 1 per cent., and additional CO_2, 0.5 per cent. found after combustion of the hydrocarbons.

We may, if we wish, alter our previous calculations for this increased weight of carbon found by the additional test. As it causes but little error to neglect it, however, this is not necessary.

Loss Due to Incomplete Combustion of Hydrogen

The method is similar to that used in the preceding calculation. The part by weight of the unburned hydrogen is 2/128 since the weight of a molecule of hydrogen is 2 atomic weights. The heat value of a pound of hydrogen is 62,100 B.t.u. The heat lost in our example is

$$\frac{2}{128} \times 62{,}100 = 970$$

B.t.u. per pound of carbon or,

$$\frac{970}{14{,}800} \times 0.837 = 0.055 \text{ or } 5.5$$

per cent. This result is approximate since the heat value of the hydrogen and its weight were used instead of those corresponding to the hydrocarbons. It is, however, closely approximate.

This shows that a small percentage of hydrogen in the flue gases causes a considerable loss of heat. To complete the combustion of this gas in the furnace it may be necessary to reduce the rate of combustion or to use a greater excess of air. If the latter is done a greater loss of heat due to this cause results. But in this as in all of the adjustments to regulate the losses a compromise is effected and conditions maintained such that all the losses sum up to the least possible.

It is useful, after making the calculations, to make a heat balance; that is, an accounting for all of the heat the coal must deliver.

Heat value equals, heat up the stack; plus CO losses; plus H_2 losses; plus heat lost by unburned carbon slipping through the grate; plus radiation from the boiler; plus heat added to the feed water.

Assuming 10 per cent. to be lost through radiation and the inefficiency of the grate, the results of our example in percentage when arranged in this manner are as follows:

One hundred per cent. equals, 15.2 per cent. (up stack); plus 8.3 per cent. (CO); plus 5.5 per cent. (H_2); plus 10 per cent.; plus 61 per cent. The last figure represents the heat utilized and the efficiency of the boiler.

Although these calculations have taken some space to explain, they will not be found difficult to perform if once understood. Enough has been presented to show their principles. It is only necessary to suggest, in closing, that when the operator thoroughly understands these principles, he need not be confined to the exact rules set down here. He may not have all of the data necessary to apply them and, if not, he can easily make estimates of what he lacks and alter the basis of his calculations so as to fit such data as he has.

Uncle Pegleg's Philosophy

The reaction jet problem is straightened out by Uncle Pegleg who clearly shows that it is "change of velocity which takes energy."

It was not until some days afterward that Uncle Pegleg came back at me about the reaction jet. You remember that he had explained how and why a turbine or reaction wheel like Fig. 1 had to run at one-half the speed of the jet to get the motion and energy all out of the water. He had also explained that a movable nozzle had to move as fast as the jet in order that the water might pay out of it like a rope and simply fall dead with no backward motion. The force with which the jet pushed on the nozzle tank in Fig. 2, as he explained it, was:

force = wt. of water discharge per sec. × vel. of jet ÷ 32 in feet per sec.

$$F = \frac{wv}{g}$$

I had evidently set him off upon a two days' think by saying that if the car with the jet on it moved at one-half the velocity of the jet the *space* through which it would move in a second would be one-half the jet velocity or $\frac{v}{2}$ and that

force × space = energy

$$\frac{w}{g} \times \frac{v}{2} = \frac{wv^2}{2g}$$

which, as he had explained before,* was all the energy that there was in the jet. It seemed then to figure that you could get the energy all out by running the nozzle at half the jet speed, while it was clear enough that the energy was not all out for the jet was still squirting backward over the ground with half the velocity that it came out of the nozzle.

One of the fellows stopped on his way to the engine room where we were talking one morning to discuss the ball game. After he had gone Uncle Pegleg asked me, "Does it make any difference whether a pitcher faces east or west?"

"No. Why?"

"Well, the earth is some 24,000 miles in circumference and it makes a turn in 24 hours. How fast is it going per second?"

Knowing that there were 5280 feet in a mile, I figured it:

$$\frac{24000 \times 5280}{24 \times 60 \times 60} = \frac{8800}{6} = 1467 \text{ ft per sec.}$$

"Well, then down at the equator a spot on the earth is moving from west to east with a speed of nearly a third of a mile a second. At the poles—no motion. In between, more or less according to latitude. Don't a pitcher ever feel the difference whether he is pitching against or with this motion, or whether he is playing in Atlanta or Boston?"

"No—what are you—stringing me?"

"Well, why shouldn't he? You've been finding out that it takes a good deal of energy to get up a velocity like that, even in a base ball."

I thought for a few minutes and then said: "Why, I don't see that the pitcher has got anything to do with that. The ball in his hand is going with the earth. *It has got that velocity already.* All he has got to do is to put that ball across the plate as hot as he can."

"That's the idea," said Uncle Pegleg. "It is *change* of velocity that takes energy. The earth is going around the sun like blazes besides spinning on its axis, but it don't bother us because we've got the motion and there's no sudden change in it. You get onto the elevator sometime with that big wrench and get one

*See page 84, May 17 number of Power.

of the boys to start you up quick, and notice how much heavier the wrench gets for an instant. It has to be gotten into motion at the speed of the elevator and your arm has got to furnish the pull to get it to moving. That's the pull that comes on you in addition to the real weight. Then let the boy stop quickly, and see how light the wrench gets. It has got the momentum of the elevator's motion. It wants to keep right on going and tends to lift itself when the elevator stops. But, and this is the important thing, it is only when the *rate of motion is being changed* that you notice anything of this kind. When the elevator settles down to running steady everything on it partakes of its motion and is just the same in the elevator as it would be on a stationary floor. When you are in a railroad car you feel the part of the car you are sitting against pushing you forward when the train starts quickly, and you have to cling to it or push backward on something to bring yourself to rest with the train when it stops suddenly; but after the motion becomes uniform you are just the same with reference to the train as you are now with reference to this room, and the movement of the train with reference to the earth wouldn't bother you any more than does the movement of the earth with reference to Mars. Two boys could toss a ball and it would be no harder or no different for the boy who tossed it toward the engine than for the one who tossed it toward the rear. So that when you make a test of a nozzle, for instance, in the hold of a vessel or on a platform car, you make it just the same as you would if you were glued to the land and you don't care whether the ship is going to Europe or to China, or whether the platform car is standing still or going like the wind, so long as its motion is uniform."

I saw then that he had got the reaction-nozzle thing straightened out.

"You take a certain amount of water," he went on, "and you put a certain amount of energy into it by making it into a jet going at a certain velocity. In order to get that energy all back you've got to bring the water back to the velocity that it had when you started. If it was at rest with reference to the earth to start with you have got to bring it back to rest with reference to the earth. If it was in a tank on a moving car you have got to bring it to rest with reference to the car. If you start with it on the moving car and put in enough energy to get it into motion with a certain velocity *with reference to the car* you don't have to take out enough energy to bring it to rest *with reference to the ground* to get your energy back, do you?"

I began to see what was coming.

"You can work from any base you want to," he said, "but you must use the same base for the finish as for the start. If you start with the water at rest with reference to the train you must bring it back to rest with reference to the train. If you start with the water at rest with reference to the ground you must bring it back to rest with reference to the ground. If you start with the water in motion with reference to your base you must subtract the energy due to its velocity from that of the jet velocity if the initial velocity were in the same direction or add it if it were in the opposite direction, for it is the *change* in velocity which requires energy.

"Now in this case (Fig. 1), you take

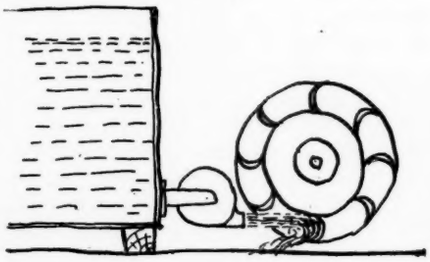


FIG. 1

water from the tank, stationary with reference to the ground, you put a certain amount of energy into it with the pump forming it into a jet. This energy will be $\frac{wv^2}{2g}$, as I explained to you the other day, where w is the weight discharged per second, v is the *change* in velocity and g is that gravity thing, about 32 feet per second. As the water starts from rest, v is the velocity of the jet. Suppose it is squirting toward the west with a velocity of a hundred feet a second and the bucket is running toward the west with a velocity of 50 feet per second. Then the jet will overtake the bucket with a velocity of 100 — 50 = 50 feet per second and the bucket will simply reverse it and throw it backward with the same speed. Well, if it is traveling west in the bucket at 50 feet a second and is thrown eastward with a speed of 50 feet a second, its speed east will just offset its speed west and it will drop dead with no motion relative to the tank where it came from.

"Now, let's take that tank on the car. Suppose the car to be running 50 feet per second eastward with reference to a

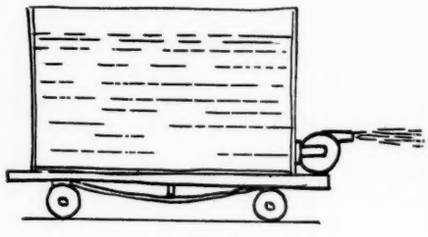


FIG. 2

fixed point on the *earth,* and the jet to be squirting westward with a velocity of 100 feet per second with reference to the *nozzle* or the *car*. If we are going to compare the energy put into the jet with the energy taken out of it we must refer the velocities from which those energies are figured to the same base. Eh? Well, let's refer them all to the earth.

"To start with, the water in the tank on the car is moving eastward with a velocity of 50 feet per second. The pump puts an amount of energy into it represented by $\frac{wv^2}{2g}$. If the car runs at one-half the velocity of the jet or through $\frac{v}{2}$ feet per second, the energy exerted upon the car will be as you figured it:

$$\text{force} \times \text{space} = \text{energy}$$

$$\frac{wv}{g} \times \frac{v}{2} = \frac{wv^2}{2g}$$

and the car will have got back all the energy that the pump put in."

"But it hasn't brought the water to rest," I remonstrated. "The water is still in motion with regard to the earth."

"So it was to start with," returned the old man. "You found it in the tank going at the velocity of the train and you leave it going at the same velocity in the other direction. The amount of energy stored in it is the same in both cases. It is the energy that was stored in it when the train was started up overcoming its inertia like the elevator overcomes the inertia of the heavy wrench. It's just the same when we get through with it as when we took it, and has nothing to do with the pump energy or jet energy that we are considering."

"Then it isn't true that a reaction nozzle like that has to run at the full velocity of the jet to get the energy all out, is it?" I asked.

"Yes, it is, because if the thing is going to run for any time it has got to take new water aboard, and the force to get it into motion has got to be charged up against the reaction, so the net force and the net energy won't figure out as you figured them. I'll tell you about that some other time."

It has been stated by Mr. Parsons that steam turbines should not be applied to vessels of lower speed than 18 knots, and the experience of the Southern Pacific with the "Creole," running between New York and New Orleans, bears this out. Originally fitted with turbines designed to drive her at 16 knots, the vessel was found to be extravagant on fuel. She was recently taken to Cramps' yard, and fitted with reciprocating engines, which, with the original boiler plant, drove the ship at 16.55 knots from port to port, the boilers steaming easily. Much interest attaches to the 16-knot Government collier, which will use turbines, with the Melvin & MacAlpine reduction gear interposed between the turbines and the propeller. Some months ago this gear was illustrated in POWER.

Steam Turbine in Sawmill Plant

By Warren O. Rogers

For the general run of sawmill plant power is usually furnished by water or by a miscellaneous collection of steam machinery. The present plant is an exception, for it is equipped with a modern 500-kilowatt turbine generating unit.

In the little lumber town of Ferguson, S. C., is located one of the largest sawmills in the South, that of the Santee Cypress Lumber Company. The steam plant consists of a cross-compound, 17 and 34 by 36-inch Buckeye steam engine, which drives the sawmill, using a steam pressure of 125 pounds per square inch. In the same engine room is a small Cameron & Barkley center-crank engine, belted to a Sprague direct-current generator of 50 kilowatts capacity. This unit is not in use at the present time, as the electric energy for lighting purposes is obtained from a turbine unit. There is also an 8x14x9x8-inch steam-driven Bury air compressor, and a three-ton Remington ammonia compressor driven by a vertical steam engine.

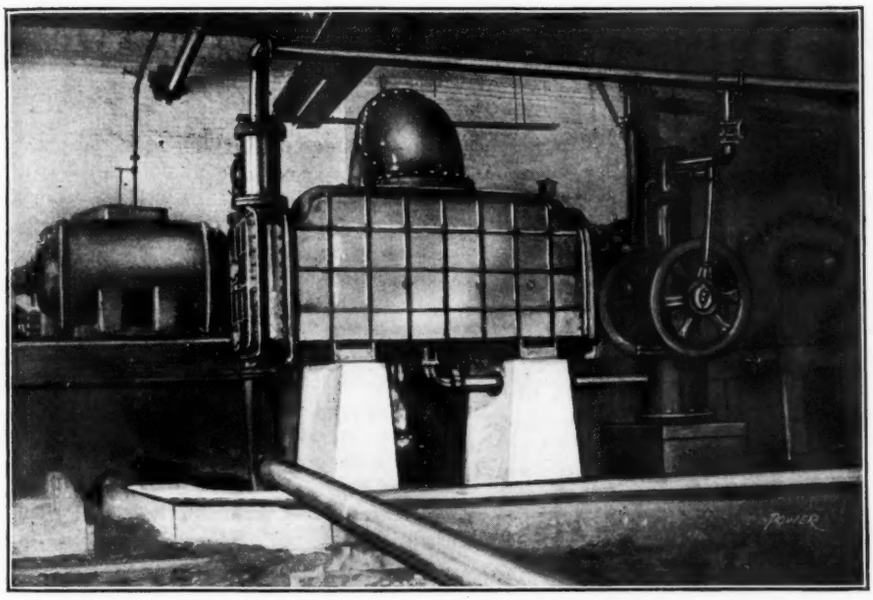


FIG. 1. ARRANGEMENT OF CONDENSING OUTFIT

Just outside of the engine room, in a new addition to the plant, is located an Allis-Chalmers turbine to which is coupled a 500-kilowatt, three-phase, 60-cycle generator.

The exciter for this unit consists of a 6x6-inch American blower engine, which is direct connected to an Allis-Chalmers direct-current 120-volt generator. There is also a motor-driven exciter set which consists of a 15½-horsepower, three-phase 60-cycle motor, direct connected to a direct-current generator designed for 120 volts.

The steam turbine is connected to a Wheeler condenser mounted on concrete piers, as shown, so that the top of the condenser rises to a hight considerably above the top of the turbine. The steam connection between the bottom of the turbine and the top of the condenser resembles in form the letter S. An Edwards steam-driven air pump is connected to the condenser, as shown in Fig. 1. The condensing water is obtained from a pump pit located at the farther end of the building. In this pit are placed a steam- and a motor-driven centrifugal pump, the steam-driven unit being held in reserve.

Fig. 2 shows a view of the top deck of the boiler room, the furnaces being of the Dutch-oven type. Two methods of supplying the boilers with fuel are employed. The bottom firing floor is arranged as in the ordinary wood-working steam plant, regulation furnace doors being supplied for the purpose of firing the larger pieces of waste, bark, etc., by hand. The upper deck receives the sawdust and finer waste material from the mill by means of the conveyer system shown in front of the boilers, and from which a spout drops down over an opening in the top of the Dutch-oven furnaces. By this method the fine wastes are handled automatically, the only firing labor in the boiler room being that of pushing the accumulated material through the firing hole into the furnace. There are eight 125-horsepower and two 250-horsepower Casey-Hedges return-tubular boilers. These boilers are equipped with metal Dutch-oven furnaces.

At present the steam-turbine unit is not loaded to its full capacity, so that it is not running at its greatest efficiency. The installation of such a prime mover, however, shows that even at a sawmill plant economy is the watchword, and that the latest development in power-plant machinery is demanded.

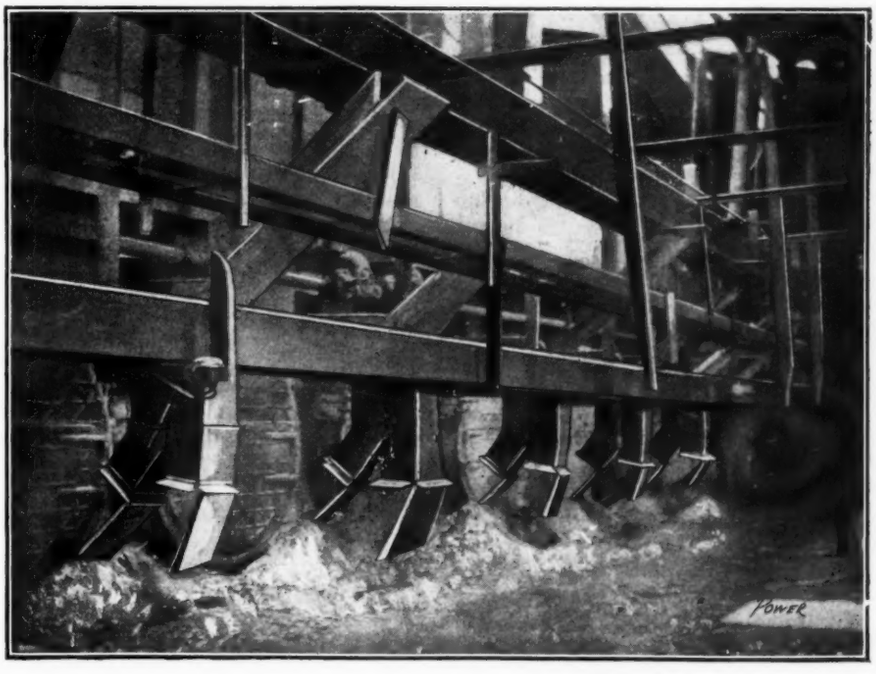


FIG. 2. METHOD OF FEEDING SAWDUST TO THE BOILER

Automatic Damper on Air Flue

By J. G. Ould

Some time ago the writer was called upon to solve a problem concerning the ventilating apparatus in a building. The method of doing the work is set forth in the hope that it may be of help to someone in a similar predicament.

The building is equipped with an indirect-heating system in which fresh air is passed over steam coils and then forced into the various rooms. There is also a suction system of ventilation. The following remarks have to do with two of the suction and one of the heating fans, all of the Sturtevant type:

The two suction fans are situated in pent houses on the roof, and are placed over large air shafts into which suction ducts are led. One fan is 60 inches in diameter, and is belted to a 20-horsepower motor; the other is 84 inches in diameter and is belted to a 25-horsepower motor. The principal work of the smaller is to exhaust the foul air from two amphitheaters used as lecture halls, while the larger draws air from various sources.

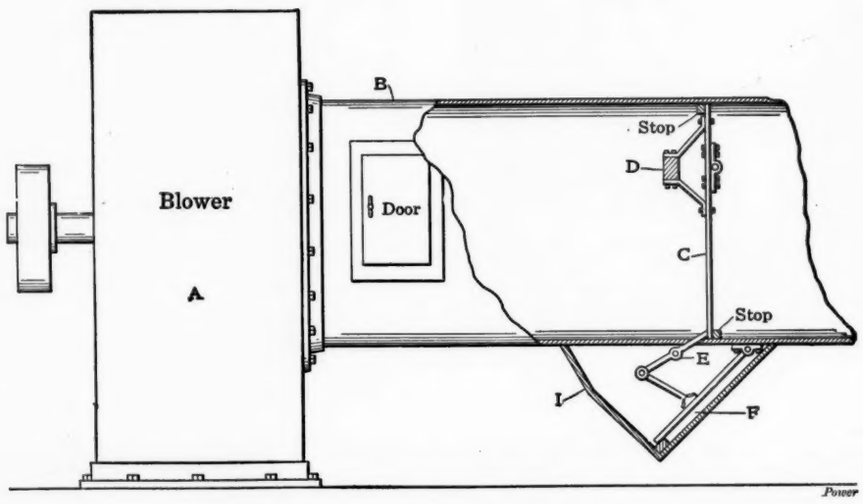
The building is of fireproofed steel construction, and has heavy concrete and tile floors, which no doubt absorb considerable heat, thereby acting as reservoirs.

The building is used in the daytime only, and consequently steam is shut off at night. This is true of the large suction fan, except that it is also shut down between noon and 1 p.m. The other suction blower is run only when the lecture halls are in use, which averages from five to seven hours per day.

When the suction fans were first put in, there were no dampers in the air shafts; so they were open to the outside air at all times. This naturally took a large amount of heat out of the building, not only in the day time, but at night as well, when the building was not in use and the fans were shut down.

The problem, then, was to install dampers in the ducts which would automatically open and shut when required; so that when the attendant started his motor, the damper would open, and when the motor was shut down, the damper would close of its own accord.

This proposition was under consideration for several days, the question being "how might it best be accomplished?" We had various sources of power at command, but finally hit upon the simple plan of using gravity in conjunction with a damper hung out of center, which proved to work very well without any attention on the part of the attendant.



AUTOMATIC DAMPER ON HEATING FAN

The damper for one exhaust fan was made of a sheet of galvanized iron 41x42 inches and hung with its shaft 6 inches off center. On the end of the shaft is a lever, which allows the damper being kept open in summer time when we wish the heat to escape. The damper is placed on the outlet side of the fan and is just heavy enough to stay shut against the natural draft of the flue, but as soon as the fan comes up to speed the damper is held open by the upward rush of air. It gradually settles back on the stops, as the fan slows down.

The other exhaust duct needed a somewhat different treatment on account of there being a large beam through the middle of the duct, which prevented the use of a large swinging damper as in the first case. This damper was made somewhat after the plan of a window shutter, except that the hinges are at the bottom of the slats instead of in the center as in the shutter. The slats are of galvanized sheet iron, with edges turned and wired, and are supported on a steel rod which acts as a shaft. These slats are connected to a rod which in turn is connected to a small lever running through the duct.

The damper in the heating duct required still different treatment. In this case the fan draws its air supply through a horizontal flue, the inlet to which is exposed to the prevailing winter winds, and, unlike the foregoing cases, the damper is on the intake side of the fan.

The style of damper used on the exhaust flues could easily be opened by a strong wind blowing into the duct, if installed on this side of the fan, and if the temperature of the incoming air happens to be below freezing, supposing the fan to be shut down and the steam cut off the heating coils, any water remaining in the coils would freeze and burst the heaters. Even if the coils did not freeze, a zero temperature brought in through a 54-inch hole would soon lower the temperature of the room.

The type of damper finally adopted to meet these conditions is shown in the illustration; *A* is the fan housing, *B* the air inlet, *C* the overhung damper, *D* the weight, *E* the latch of the lock, and *F* the piston which actuates the latch to unlock the damper. The damper is shown closed and locked.

Its operation is as follows: As soon as the fan speeds up, it exhausts the air in the duct between the damper and the rotor and the outside air pressure forces up the piston *F*. This unlocks the damper, which with the suction on one side and the current of air on the other, begins to swing about its shaft. The overhung weight *D* then comes into play and with the aid of the air current keeps the damper steady and horizontal.

When the fan stops, the damper slowly returns to its original position, the weight *D* now helping to close it and force it over the latch, which as soon as the damper passes, springs up into place and locks.

This scheme has worked automatically for about four years without giving trouble of any kind.

Two reinforced concrete tanks have been built as part of a concrete-block chimney in France. The chimney is 180 feet high, the diameter at the summit 7 feet 9 inches and the tank capacity 33,000 gallons.

Mechanical Refrigeration

By F. E. Matthews

An elementary treatment of commercial systems with simple illustrations to show the principles of operation.

Practical mechanical refrigeration may be said to date back to 1855, in which year the ammonia-absorption machine was invented by Carré, and the Perkins ether compression machine, patented in 1834, was simultaneously commercialized by two different inventors in two different countries—Professor Twining in America and Mr. Harrison in Australia. It is a rather strange coincidence that the development of the two systems so commonly employed today, i.e., the absorption and the compression systems, should have been commenced the same year. Generally speaking, while both the absorption and the compression systems have been employed in connection with both the direct expansion and the brine systems for the production of both medium and extremely low temperatures,

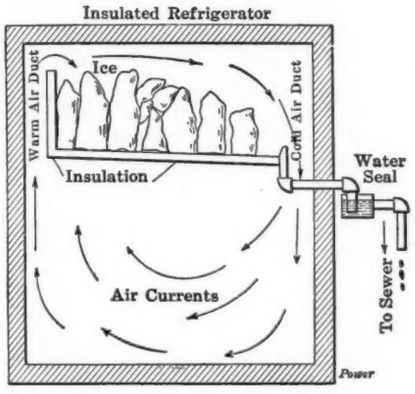


FIG. 1. REFRIGERATOR COOLED BY ICE BUNKER

brine has been used more commonly in connection with the absorption machine and direct expansion in connection with the compression machine. The former has been employed more frequently for extremely low and the latter for comparatively low temperatures. The absorption machine is generally conceded to be more economical than the compression machine when producing extremely low temperatures, under which condition, because of the lightness of the gas under the correspondingly low pressures, the compression machine loses greatly in both efficiency and capacity. At the present time, however, it is problematical whether the substitution of the higher-efficiency gas engine as a prime mover for the inherently low-efficiency steam engine will not put the compression system in position to produce as economical results as the absorption system, even under the disadvantage of low temperatures. Since the compression system is in far more common use today, and since a large number of the parts of the two systems are common, the details of construction of the compression machine will first be considered.

The function of a refrigerating apparatus, whether it be an absorption or a compression refrigerating machine, or simply a bunker full of ice, is to provide a heat-absorbing medium which, after it has absorbed its fill of heat from the products to be cooled in the cold-storage rooms, may be removed from the coolers so that the heat absorbed may also be removed. After its removal from the coolers, this medium may be divested of its heat, after which it may be allowed to return to the coolers to absorb more heat, as in the case of ammonia or brine circulated through coolers; or it may be thrown away and a new supply introduced, as in the case of cooling by the use of ice.

A diagram of the simplest arrangement of an ice bunker for cooling a cold-storage compartment is shown in Fig. 1. The ice in this case is stored on a water-tight floor over the compartment to be cooled, the cooling being effected by a natural circulation of air up over the ice and down through the cold-storage compartment. Movement of air without the use of a fan driven by an outside source of power is comparatively slow, owing to the limited power available in the difference in weight of the ascending column of warm and the descending column of cold air. Since both the bunker and the room below are completely filled with air, the sinking of air in one place displaces air in another place. The warmer air is forced up to the bunker where it in turn becomes gradually cooled, finally flowing down by gravity into the room below.

Bunkers for ice, or other cooling means, are usually constructed in the form of a tank with one side removed, as shown in Fig. 1, ample spaces for ducts being allowed between the sides of the tank and the sides of the room to permit the air to circulate freely. Were the left side of the tank also removed, it is obvious that the tendency would be for the cold air to flow off of the ice in both directions. This would give rise to conflicting currents of air which would check the circulation, and for the same rate of air circulation would require a greater difference in temperature between the air in the bunker and that of the cooler below.

To prevent the cooling of the rising column of air in the warm air ducts as well as that lying next to the floor in the bunker, which not only retards circulation by reducing the difference in temperature, but may also precipitate moisture when the warm air approaches saturation, the air-duct walls and bunker floors should be insulated. When operating with properly insulated bunkers and liberal ducts, the circulation should be sufficiently rapid to carry any moisture-saturated air that may enter the cooler from the outside or that may have become saturated through contact with stored products to the ice chamber before its moisture can be precipitated by contact with the cold surfaces of the cooler. When cooled in this way the excess moisture is precipitated in the ice bunker and flows away with the melted ice. Where the warm air ducts or the bunker floors are not insulated, excess dampness can be prevented only by the use of some such deliquescent salt as calcium chloride. This is the case when the air of the cooler is not allowed to come in contact with the melting ice but is chilled by contact with the metallic floors of overhead bunkers or the walls of tank bunkers.

Where salt is used so that the resulting temperatures are below freezing, the precipitated moisture is frozen on the heat-absorbing surfaces and this difficulty is avoided, but in time this accumulation has to be removed, thus giving rise to other difficulties.

Exceptionally well insulated coolers

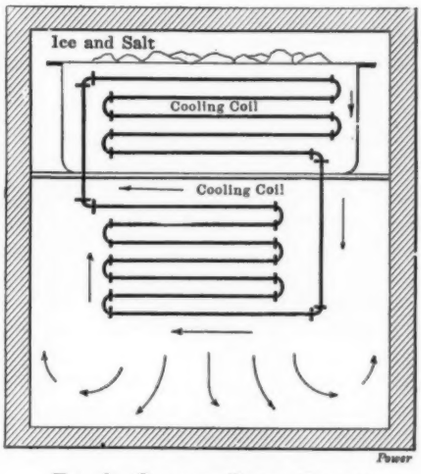


FIG. 2. GRAVITY BRINE SYSTEM

have been constructed in which it is possible to maintain a temperature as low as from 38 to 40 degrees Fahrenheit in compartments cooled by natural circulation and ice without the use of salt. The average cooler, however, is not capable of producing such favorable temperatures, and the present-day demand for lower temperatures makes it neces-

sary to resort to other means. The addition of salt allows the natural circulation and ice system to satisfy a few such requirements, and the further addition of a fan to force the circulation may allow it to include a few more. For widening the application of ice cooling, the system illustrated in Fig. 2 has been devised. This system consists essentially of a tank for holding the ice and salt in the proportion required to produce the desired temperatures, and a continuous pipe circuit, a part of which is located in the cold-storage compartment where it absorbs heat and the other part in the ice tank where it gives up the heat to the ice or freezing mixture of ice and salt, as the case may be. The pipe is filled with brine of just sufficient density to insure against freezing. The brine in the coil of pipe located in the tank, becoming heavier, flows down into the coil located in the cold-storage compartment, causing the warmer, lighter brine to pass upward and take its place in the ice tank. Heat is conveyed from the cold-storage compartment to the ice chamber by the natural circulation of the conveying medium as in the preceding case, but in this case the medium is a liquid while in the preceding one it was a gas. An abnormal rise in temperature of the cooler increases the velocity of the brine circulation and consequently increases the refrigerating capacity of the system. By the use of this system it is possible to prevent the air of the cold-storage compartment from becoming contaminated by contact with melting ice and unsanitary ice bunkers.

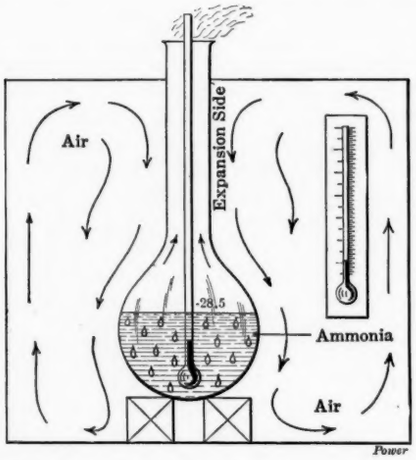


FIG. 3. ILLUSTRATING DIRECT-EXPANSION SYSTEM

The process of mechanical refrigeration when the refrigerating medium is a condensable gas, or more accurately speaking, a liquid having a sufficiently low boiling point under atmospheric or other conveniently produced pressures, to produce the desired temperatures, may be explained as follows:

Fig. 3 represents a flask partly filled with a refrigerant such as anhydrous ammonia. Since the pressure on the refrigerant in the open flask is only that of the atmosphere, it will boil at —28.5 degrees Fahrenheit. As this temperature is far below that of the surrounding air under usual conditions, heat will pass from the air into the refrigerant; the former would be refrigerated and the latter heated up to the boiling point if it were not already boiling. At the boiling point the absorption of a definite amount of heat from the air effects the evaporation of a definite amount of the liquid; the anhydrous ammonia, absorbing heat directly through the walls of the containing vessel from the surrounding air, operates just as it and similar refrigerating media do in the direct-expansion refrigerating system. This cooling effect is hastened by a marked circulation of air around the flask, occasioned by the greater specific gravity or weight of the cooled film of air lying next to the flask, which flows down and away at the bottom of the flask, allowing the warmer air to rise and take its place at the top.

The brine system has its elementary counterpart in such an arrangement as that illustrated in Fig. 4, in which the flask containing the boiling anhydrous ammonia is immersed in a second containing vessel, such as a large beaker, filled with brine, a solution of sodium chloride (NaCl), calcium chloride (CaCl), or other salt in water. The low first cost, together with certain physical and chemical characteristics, has practically limited the commercial brine system to the use of either sodium or calcium chloride solutions. As in the preceding case, the boiling refrigerating medium absorbs heat from the surrounding medium, in this case brine which we will assume is of such a nature as not to freeze at —28.5 Fahrenheit. Since this solution does not freeze at the temperature of the evaporating refrigerant, and the latent heat of fusion of the liquid is not extracted, an evaporation of only a comparatively small amount of the refrigerant suffices to cool the surrounding solution almost down to —28.5 Fahrenheit. This cooling effect is hastened by a marked circulation within the solution itself, set up by the difference in specific gravity of the colder part of the liquid lying next to the flask, which sinks to the bottom of the beaker, allowing the warmer part to flow in and take its place.

The brine in the beaker having become colder than the surrounding air, heat flows from the air to the brine, just as in the preceding case it flowed from the air to the ammonia. It will be noted, however, that difference in temperature, and accordingly the flow of heat, between the air and the brine can never be so rapid as that between the air and the ammonia, except in the limiting case in which the brine and the ammonia are of the same temperature, when unfortunately the ammonia would have no cooling effect on the brine. If the same amount of the same refrigerant be placed in flasks of the same shape and size in both the foregoing experiments, it will be found that the brine employed in the second case will assume a temperature intermediate between that of the boiling ammonia and that of the surrounding air, and were there any way to measure the cooling effect produced on the air, it would probably be found that less heat would be absorbed in the latter than in the former case, notwithstanding the fact that the heat-absorbing surface of the beaker is greater than that of the flask. A very crude idea of this difference might be gained by surrounding the vessels used in each experiment by boxes made of the same size and material and noting the temperature of the air within.

The flasks containing the ammonia employed in the above experiments correspond to the expansion pipes immersed in and employed for the purpose of cooling the brine in the brine system described later, and the beaker containing the brine to the brine pipes immersed in and employed for cooling the air in the cold-storage compartments. If the flask of ammonia employed in the second experiment had been immersed in water, as shown in Fig. 5, instead of brine, as shown in Fig. 4, the water which freezes under atmospheric pressure at 32 degrees Fahrenheit, will give up first its specific then its latent heat to the anhydrous ammonia boiling under atmospheric pressure at —28.5 Fahrenheit, with the result that the water will first be cooled from its initial temperature to 32 degrees Fahrenheit, then be frozen at 32 degrees. If sufficient ammonia still remains and heat is not absorbed too rapidly from the surrounding air, the ice will be cooled several degrees lower than 32 degrees Fahrenheit; in fact, that part lying nearest the flask may be chilled to the temperature of the evaporating liquid, —28.5 degrees Fahrenheit. If a pound of anhy-

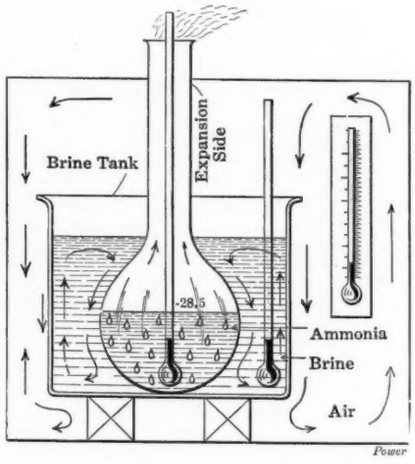


FIG. 4. BRINE CIRCULATING SYSTEM

drous ammonia be evaporated from and at —28.5 degrees Fahrenheit in the flask, the amount of heat which it will absorb will be 555 B.t.u., which is its latent heat of vaporization. If now it be assumed that this heat is all drawn from three pounds of water at an initial temperature of 73 degrees Fahrenheit, it will just suf-

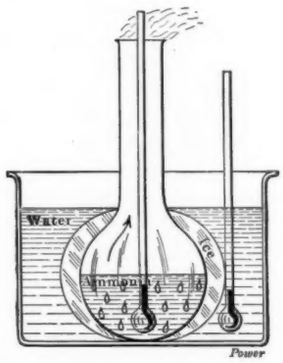


FIG. 5. ELEMENTARY PLATE-ICE-MAKING SYSTEM

fice, assuming no losses, to cool it down to the freezing point and then to freeze it at 32 degrees Fahrenheit, 42 B.t.u. for each of the three pounds being required to overcome the specific heat of the water in cooling it through 41 degrees, and 144 B.t.u. for each pound of water at 32 degrees Fahrenheit being required to overcome the latent heat of fusion.

Three pounds water × drop in temperature from 73 to 32 degrees × specific heat of water (unity) + three pounds water × latent heat of fusion of water (144 B.t.u.) = latent heat of vaporization of the ammonia (555 B.t.u.) required to cool and freeze the three pounds of water.

$$3\times1(73-32)+3\times144=555.$$

If carried out in an ice plant instead of a laboratory, the apparatus requisite for the above experiments would probably appear more nearly as in Fig. 6. In this illustration the glass flask is replaced by an iron flask, or drum, such as is commonly used for shipping ammonia, carbon dioxide and other liquefied gases. The liquid from this flask is allowed to "expand" or escape in a small stream through a valve V and a pipe leading to an expansion coil. After traversing this coil, the vaporized ammonia escapes to the atmosphere. This outlet, however, is provided with a valve V_1, which, when closed, diverts the ammonia through a second pipe connecting through two outlets, each provided with a valve, the one leading to a tank of water and the other to an ammonia compressor. If diverted into the water, the ammonia vapor will be absorbed and be recoverable in the form of aqua ammonia, a principle which is employed in the absorption type of refrigerating machine. The ammonia vapor may be conducted to an ammonia compressor and suitable condenser, where it will be condensed and recovered in the form of anhydrous ammonia, a principle which is employed in the compression type of refrigerating system. The several parts used in commercial refrigerating systems to take the place of those illustrated in Figs. 3, 4 and 5 are represented diagrammatically in the figures that are to follow. Fig. 7 is such a conventional representation of an ammonia flask or receiver, a controlling or expan-

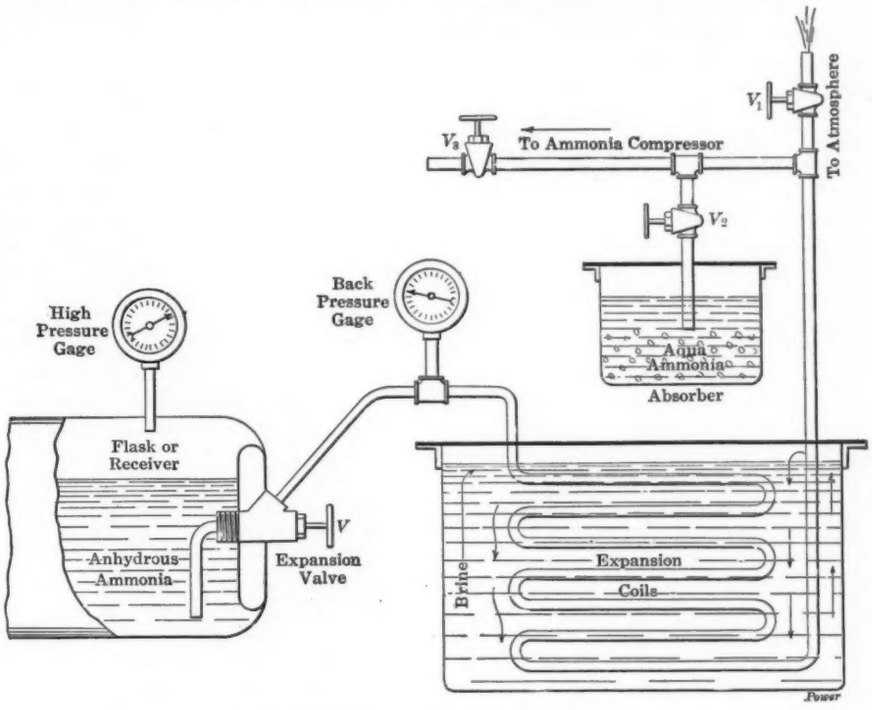


FIG. 6. ELEMENTARY BRINE SYSTEM

sion valve V and a container of the boiling liquid ammonia, or expansion coil.

Were it not for the initial cost of the refrigerating medium, this elementary refrigerating system in which the ammonia is allowed to escape to the atmosphere after evaporation might find commercial application, but since anhydrous ammonia commands a practically fixed market price of $0.25 and upward per pound, according to the distance from point of production, such a system would be eminently impracticable.

A commercial compression refrigerating system consists of a set of pipes, or other containing vessels, in the cooler, in which the refrigerating medium absorbs heat at a low temperature from the products to be refrigerated; a second set of pipes, or other containing vessel, outside of the cooler in which the refrigerating medium gives up its heat to a secondary cooling medium, such as water or air, at a comparatively high temperature, and a compressor.

The pipes, located in the cooler, through the walls of which the ammonia absorbs heat from the objects to be refrigerated, are erroneously called expansion coils, and those through the walls of which the ammonia gives up its heat to the natural cooling medium, water, condenser coils.

The compressor is a mechanical means of circulating a heat-absorbing medium through the cooler, and of removing from this medium the heat absorbed *en route*. Heat is absorbed in the cooler from the atmosphere, products, etc., by virtue of the fact that the refrigerating medium is lower in temperature. This same heat must be similarly removed from the refrigerating medium by introducing a secondary medium materially lower in temperature. Since there is no secondary cooling medium available of a lower temperature than the refrigerating medium, even at the temperature at which it returns from the cooler after having absorbed large quantities of heat, it becomes necessary to raise the thermal level of the heat in the refrigerating medium sufficiently to allow it to gravitate

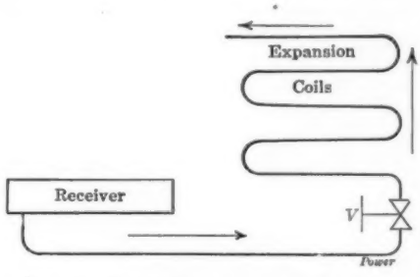


FIG. 7. CONVENTIONALIZED DIAGRAM OF FIG. 6

into the cooling medium. This is effected in the absorption machine by the direct application of heat in the "generator," and in the compression system by the performance of work in the "compressor."

In the operation of a compression system, almost any gaseous working medium might be employed. In practice, how-

ever, the list is limited to only such gases as are capable of being liquefied under ordinary natural temperatures and not too high mechanically produced pressures. Judging from the relative number of commercial installations employing the different media, one may assume that under the average conditions anhydrous ammonia comes nearer to fulfilling all the requirements of a practical working medium than any other.

The system previously described and illustrated in Fig. 3 consists of a single member, i.e., a containing vessel for the working medium. The refrigerating process might have been carried out just as well had there been no outer containing vessel and no second liquid. In that case the heat would pass from the air surrounding the test tube, directly to the ammonia, just as the air of a cold-storage compartment is cooled by "direct expansion," a system which is differentiated from the brine system by the location of the "expansion coils" containing the boiling or expanding ammonia, in direct contact with the atmosphere of the cold-storage rooms.

The commercial system corresponding to the one illustrated in Fig. 4, in which the evaporation of the ammonia cools the surrounding water, which in turn cools the surrounding air, is the "brine system," which takes its name from the fact that the cooling effect of the refrigerating medium is expended on a more or less uncongealable solution, such as sodium chloride (NaCl) or calcium chloride (CaCl) brine, in which the cooling solution circulated through a secondary system of coils in the cold-storage compartments is made the vehicle for conveying the refrigeration to, or, more properly speaking, the heat from, the atmosphere in the cold-storage rooms.

In order to make the systems illustrated in Figs. 3, 4, 5 and 6 commercially practicable, means must be provided for converting the gasified anhydrous ammonia back into the liquid state. In other words, after the ammonia has evaporated in the expansion pipe coils and has absorbed its full complement of heat, much as the sponge sucks up its fill of water, the heat and water with which the ammonia and a sponge are respectively charged must be squeezed out before they can again perform the function of absorption. In the case of the sponge, this is accomplished by the simple application of pressure. In the case of ammonia, pressure must not only be applied, but, since heat can be made to flow only from a relatively warmer to a relatively cooler substance, some cooling medium must be provided as well. The two cheapest and otherwise most convenient natural cooling media to be obtained are air and water. On account of the low specific heat of the former and the fact that it is usually at a higher temperature than water found in the same locality, we are practically limited to the use of water.

Before the ammonia can liquefy, all of the heat absorbed in the cold-storage rooms must be squeezed out, or rather be induced to flow out of the ammonia into the cooling water. From this it appears that were the water sufficiently cold, i.e., colder than the gaseous ammonia returning from the cooler, the flow would take place without the increase in pressure and temperature, and there would be no need for a compressor. On the other hand, if so cold a medium were available, it would be used directly for absorbing heat in the place of the ammonia or the ammonia and brine, as it would then be sufficiently cold to induce a flow of heat from the products to be refrigerated.

Since the pressure and consequent temperature of the cold ammonia gas returning from the expansion pipes must be raised before a heat transfer can be made to take place between it and cooling water at ordinary temperatures, the use of a compressor or suitable gas pump, in the case of the compression system, and some other means of heating and increasing the pressure of the refrigerating medium, in case of the absorption system, must be employed. In either system there must also be provided a suitable cooling chamber or condenser in which the cooling water can be brought in sufficiently close proximity to the refrigerating medium to allow the necessary heat flow from the hot ammonia to the cold water to take place.

Fire and Water Tube Boilers

By William Westerfield

Notwithstanding claims to the contrary, soot does not affect the efficiency of the fire-tube more than that of the water-tube boiler. Fire-tube boilers are less prone to prime, and, other things being equal, they are as efficient as the water-tube type.

In this article it is not the writer's purpose to speak in condemnation of any class of equipment, but to call attention to the fact that the representations made in favor of any class of power-plant equipment, or any other for that matter, by those engaged in the building or selling of it, must not be given too much weight. The same human frailty that leads the ordinary worker to imagine himself to be worth more than he really is, leads the builders and sellers of equipment to believe in the merits of their

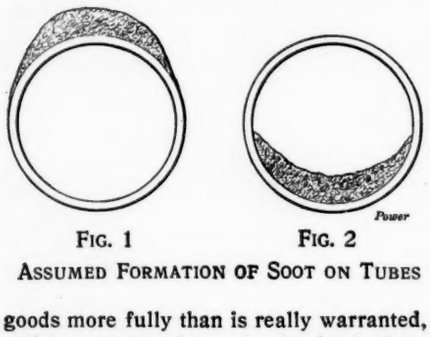


FIG. 1 FIG. 2

ASSUMED FORMATION OF SOOT ON TUBES

goods more fully than is really warranted, and to overstate them. It also leads them to see theoretical advantages, and to overlook practical disadvantages in the things they are building or selling. The situation in which they are placed, as advocates for something in which they have a lively personal interest, naturally leads them to take a more or less one-sided view, so that they are prone to give all the facts favorable to their output, but omit all that are not favorable. This treatment, when it is taken "as directed" makes a strong case for a weak proposition.

I have in mind a claim made for the water-tube boiler, which in theory seems all right, but which in practice does not stand up. It is claimed that in the fire-tube boiler the soot accumulating in the tubes has a greater insulating effect than it has in the case of the water-tube boiler. This is usually illustrated as in Figs. 1 and 2. In Fig. 1 is shown the tube of a water-tube boiler, with the accumulation of soot and ashes on the top of the tube, where, according to the claim, it does not interfere with the draft as it would in the fire-tube boiler. This might be true, but as a matter of fact it is not. In the first place, there is no excuse for allowing the tubes in any type of boiler to get into the condition shown. It may be that boilers are run that way in some "Arkansaw" sawmill plant of 10-horsepower capacity, but in any plant operated by an engineer, never. Fig. 2 is employed to show how the accumulation forms in the fire tube, showing a tube half full of ashes. If this condition is permitted, then we must admit that the draft will be seriously interfered with, and the steaming capacity of the boiler considerably reduced, no matter of what type it may be.

But these illustrations fail to illustrate, as well as the descriptions fail to describe the actual conditions. Any engineer knows that soot does not hunt for the top

or bottom of a surface to the extent indicated. On the contrary it sticks wherever it lands, whether it is top or bottom. If you open the smoke-box door of a fire-tube boiler you will find that up to a certain point the accumulation will be of almost uniform thickness all around the inner surface of the tube. When it is allowed to accumulate, say, for a week, it is likely that the mass would fall to the bottom surfaces, and fill up the tubes as indicated. But why take an abnormal

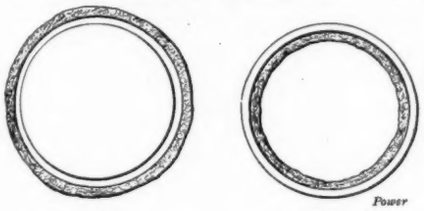


FIG. 3. FIG. 4

ACTUAL FORMATION OF SOOT ON TUBES

condition for the purpose of illustration, when it is not likely that condition will exist in one case in a million? And for the same reason, the soot accumulation does not form on the top of the water tube and leave the lower surfaces clean as indicated, but forms almost evenly all over its outside surface.

The manner in which the accumulations really form is shown in Figs. 3 and 4 respectively. The fire-tube boiler has the advantage of being easily reached in every part of the tube and shell surfaces, inside and outside. The scraper used for the inside of the tubes of fire-tube boilers is familiar to all engineers and firemen. For the outside of the tubes the writer has used scrapers of the form shown in Fig. 5. These are made of two lengths for convenience, the shorter one usually about 12 and the other about 18 inches long, and by their use all parts of the outside surfaces of the tubes may be reached and cleaned easily and thoroughly. The inside edge of the curved part of this scraper is made sharp in order to enter the scale more readily and remove it. These may be made in any shop where there is a forge and hammer.

In order to get good results with any type of boiler, it is imperative that the heating surfaces be kept clean, inside and outside. When it is understood how easy it is for the firemen to blow the soot and ashes from the boiler flues or tubes with the apparatus now furnished for that purpose, no excuse can be found for allowing them to fill up, or to accumulate sufficient soot and ashes seriously to affect the steaming of the boiler. The writer has never seen a water-tube boiler that could be cleaned in this manner as quickly and as easily as the old return-tube or marine boiler of the Continental type. In five minutes the fireman can open the smoke-box doors and blow the accumulation from every tube in a large boiler of either of the above types, and this repeated twice in twelve hours will keep the tubes in very good condition in so far as soot is concerned. But he has yet to see the water-tube boiler that can be cared for in this respect so quickly.

In the fire-tube boiler the blower is used from the front, where plenty of working room must be provided for other reasons; hence no extra room need be provided to operate the cleaning apparatus. In the water-tube boiler settings, the openings are usually located in the sides, as well as front and rear, so that it is necessary to allow considerable space between settings to enable the men to get between them when using the blower.

In Doctor Bell's excellent work on "Electric Power Transmission," on page 330 there is a table of recorded boiler performances. The best performance with any fuel is stated to be 13.12 pounds of water evaporated per pound of fuel, for a return-tube, and 13.01 for a water-tube boiler. On Cumberland coal the performance of the return-tube boiler was 12.07 pounds evaporation per pound of fuel, and that of the water-tube boiler, 10.79 pounds per pound of fuel. Quite likely there are other records that show the water-tube boiler in a better light, but the preceding ought to go far toward convincing anyone that the race of the "old reliable" that our forefathers got along with is not fully run.

In the writer's opinion, the water-tube boiler is especially to be recommended for large installations and for high pressures. For medium-size and small power plants, and for pressures up to, say, 150 pounds, the writer prefers the fire-tube boiler. They are more easily cleaned, can be cared for by men of less skill than

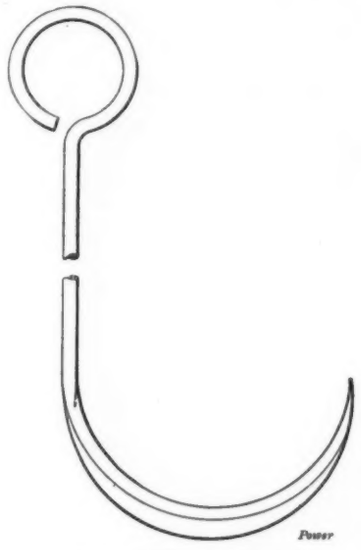


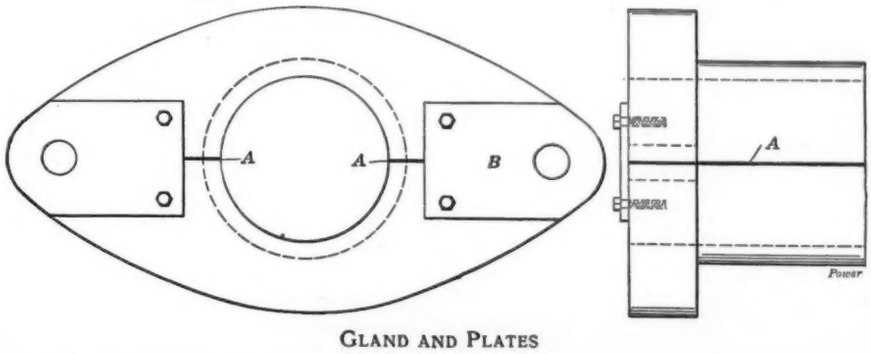
FIG. 5. TUBE SCRAPER

is required for the water-tube boiler, and are less liable to give trouble when unskilfully handled. They are not so likely to prime when forced, due to the fact that having a larger relative volume of water from which to generate steam, the ebullition is not so active, hence there is less tendency to throw up spray to become entrained in the steam current to the outlet. This is one great advantage in itself. In the water-tube boiler, the ebullition being very active and the steam bubbles being formed so rapidly and filling so much of the water space in the boiler, the water level is caused to rise abnormally high when the boiler is being forced, and as soon as the forcing ceases the level will fall abnormally low.

Split Stuffing Box Gland

While looking around a power plant in Chattanooga, Tenn., the writer noticed the method, illustrated herewith, of making a split gland in a case where it was found necessary to devise some method of permitting the glands to be taken from the piston rod of a pump without taking the pump apart.

The gland was of the ordinary type sawed through the center lengthwise, as shown at *A*. In order to secure the two sections after being placed on the piston rod, two iron reinforced pieces were made and drilled as shown at *B B*. The larger holes were for the passage of the gland studs, and the two smaller holes for the two bolts which screw into holes drilled and tapped in the face of the gland. By removing two bolts and the stud nuts, the gland can be easily removed.



GLAND AND PLATES

Electrical Department

Installing Induction Motors

By R. H. Fenkhausen

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

The Connecting Wires

When considering the installation of an electric motor, the first problem encountered is the calculation of the supply circuit. In calculating the proper size of wire to use for a given induction-motor installation, a more careful consideration of the factors involved is necessary than would be required in a direct-current installation.

In the first place the relation of the supply voltage to the motor voltage requires careful study, because the torque developed by any induction motor is proportional to the square of the voltage and a very small difference in voltage results in a marked difference in the torque of the motor, especially at starting. Most commercial induction motors will operate on voltages within 10 per cent. of their rating, but with impaired operating characteristics. To enable a closer selection of voltage one large manufacturer puts out two lines of motors, one for 200-400 volts and one for 220-440 volts, which allows the choice of a motor suitable for voltages of 180-240 and 360-480. When the voltage of the line is below the motor voltage the wiring to the motor must be calculated for as small a drop as practicable in order that the available voltage will not be greatly reduced by excessive line losses. If the line voltage is higher than the motor voltage a drop of 5 to 10 per cent. is permissible in the wiring and results merely in the loss of that much power, which may be offset by the saving in copper, especially for large motors or long distances.

The power factor of an underloaded motor is improved by a reduction in voltage, which fact should be considered when installing a motor large enough for future extensions. A three-phase motor with its windings normally delta connected may be connected star- or Y-fashion until the load increases to a value requiring the full horsepower, and an improvement in power factor secured. The torque will, of course, be reduced but it will be sufficient for the light load.

For induction motors not exceeding 25 horsepower in capacity, the familiar direct-current formula,

TABLE 1. FULL LOAD CURRENTS PER PHASE.

Average Power Factor and Efficiency.

	110 Volts.			220 Volts.			440 Volts.			550 Volts.			
H.P.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	H.P.
2	23.80	10.10	11.54	11.90	5.05	5.77	5.95	2.52	2.89	4.76	2.02	2.30	2
5	52.30	22.92	26.50	26.15	11.46	13.25	13.07	5.73	6.63	10.46	4.58	5.30	5
7½	68.75	34.42	33.70	34.37	17.21	19.85	17.19	8.60	9.93	13.75	6.88	7.94	7½
10	90.60	45.30	52.40	45.30	22.65	26.20	22.65	11.32	13.10	18.12	9.06	10.48	10
15	132.80	66.40	76.80	66.40	33.20	38.40	33.20	16.60	19.20	26.56	13.28	15.36	15
20	175.20	87.40	101.30	87.60	43.70	50.70	43.80	21.85	25.35	35.04	17.52	20.26	20
25	219.00	109.60	126.70	109.50	54.80	63.40	54.70	27.40	31.70	43.80	21.90	25.34	25
35	321.00	160.50	185.80	160.50	80.20	92.90	80.00	40.10	46.40	64.20	32.10	37.15	35
50	428.00	214.00	247.20	214.00	107.00	123.60	107.00	53.50	61.80	85.60	42.80	49.44	50
75	656.00	328.00	379.10	328.00	164.00	189.50	164.00	82.00	94.70	131.20	65.60	75.82	75
100	848.00	424.00	487.20	424.00	212.10	244.10	212.20	106.10	123.40	169.60	84.80	97.44	100
125	1067.00	533.50	613.90	533.50	266.90	307.50	266.90	133.40	155.10	213.40	106.70	122.78	125
150	1300.00	650.00	751.00	650.00	325.00	376.00	325.00	162.50	187.40	260.00	130.00	150.20	150
200	1696.00	848.00	974.40	848.00	424.00	487.20	424.00	212.00	244.00	339.20	169.60	194.80	200
250	2134.00	1067.00	1228.00	1067.00	533.50	613.90	533.50	266.90	307.50	426.80	213.40	254.60	250
300	2600.00	1300.00	1502.00	1300.00	650.00	751.00	650.00	325.00	376.00	520.00	260.00	300.40	300

TABLE 2. MOTOR WIRING.

For a three-phase motor, use the current per wire of a corresponding two-phase motor, and determine the size of wire as though for that motor.

Approximate Loss in Per Cent., at Different Standard Voltages.				Current per Wire × Distance in Feet for Ten Standard Wire Sizes.									
110 V.	220 V.	440 V.	550 V.	No. 0.	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.	No. 6.	No. 8.	No. 10.	No. 12.
1%				5,300	4,200	3,320	2,630	2,090	1,650	1,310	825	520	325
2	1%			10,600	8,400	6,640	5,260	4,180	3,300	2,620	1,650	1,040	650
3	1½			15,900	12,600	9,960	7,890	6,270	4,950	3,930	2,475	1,560	975
4	2	1%		21,200	16,800	13,280	10,520	8,360	6,600	5,240	3,300	2,080	1,300
5	2½	1½	1%	26,500	21,000	16,600	13,150	10,450	8,250	6,550	4,125	2,600	1,625
6	3	1½	1½	31,800	25,200	19,900	15,780	12,540	9,900	7,860	4,950	3,120	1,950
8	4	2	1¾	42,400	33,600	26,560	21,000	16,700	13,200	10,480	6,600	4,160	2,600
10	5	2½	2	53,000	43,500	33,200	26,300	20,900	16,500	13,100	8,250	5,200	3,250
	6	3	2¾	63,600	51,400	39,840	31,550	25,080	19,800	15,720	9,906	6,240	3,900
	8	4	3¼	84,800	67,200	53,120	42,000	33,400	26,400	20,960	13,200	8 320	5,200
	10	5	4	106,000	84,000	66,400	52,600	41,800	33,000	26,200	16,500	10,400	6,500
		6¼	5	132,500	105,000	83,000	65,750	50,200	41,200	32,750	20,625	13,000	8,125
		8	6½	169,600	134,400	106,240	84,000	66,800	52,800	41,900	26,400	16,640	10,400
		10	8	212,000	168,000	132,800	105,200	83,600	66,000	52,400	33,000	20,800	13,000
			10	265,000	210,000	166,000	131,500	104,50	82,500	65,500	41,250	26,000	16,250

$$\frac{22\,I\,D}{V} = c.m.$$

may be used and the result increased by one size of the Brown & Sharpe gage to cover the inductive drop, etc. In this formula $c.m.$ = circular mils of wire cross-section; I = amperes per phase at full load; D = length (in feet) of one

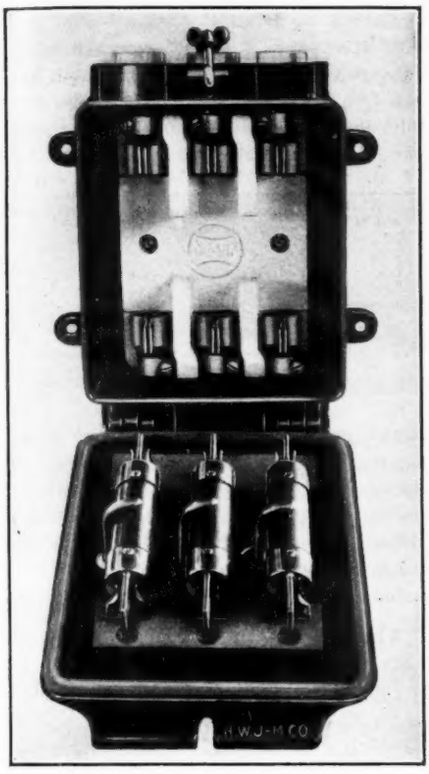


FIG. 1. INCLOSED FUSE BOX

wire, V = the drop in volts, and 22 is the resistance per mil foot of a two-wire circuit, allowing for sag or other irregularities.

The full-load current in amperes is usually given on the motor name plate, but may be taken from Table 1. This should be increased by 50 per cent. to allow for overloads, before applying the formula. Table 2 will serve to reduce the calculation required in most cases. This gives the relation between the distance × amperes per wire and the approximate loss in percentage of the voltage at the motor, for wires of standard sizes from No. 0 to No. 10.

For example, suppose a 15-horsepower two-phase motor is to be located 150 feet from the nearest point on a 220-volt local main, and it is desired to keep the loss down to 3 per cent. maximum, because the voltage is not always right up to normal. From Table 1 the full-load current is 33.2 amperes per wire, and adding 50 per cent. for overloads gives practically 50 amperes maximum. The distance × current = 150 × 50 = 7500. Starting at 3 per cent. in the 220-volt column of Table 2, follow that line into the right-hand section of the table until the nearest number to 7500 is found; that is 7875, and it is in the No. 6 column, so that No. 6 is the size of wire that would be used for direct current. Taking one size larger, to allow for inductive drop, gives No. 5 as the size to be used in the case assumed.

The value of the secondary current for multiple-speed motors should be obtained from the manufacturers of the motor before attempting to install the secondary wiring between the motor and the controller.

For very large feeders supplying a large number of motors, it is preferable to use several small cables in parallel rather than one large cable, because the inductive drop in one large wire supplying a load of low power factor is considerably greater than that in several smaller wires of equal total cross-section. This feature does not become important, however, until the size of the large conductor is greater than No. 00 or 000 Brown & Sharpe gage. The inductive drop in stranded cables is greater than that in solid wire of equivalent cross-section, due to their greater diameter, and also to the helical twist given the stranding. The difference is so small, however, that it is negligible except for long transmission lines.

In alternating-current conduit work all the wires of a circuit must be in one pipe, or else the drop will be prohibitive and heating of the conduit will result. For open work the different wires of a circuit should be located as close together as circumstances and voltage will permit, as the inductive effect is lessened by reducing the distance between the wires.

OVERLOAD PROTECTION

Nearly all modern motor-starters for use with squirrel-cage induction motors are provided with two sets of taps, one for running and one for starting.

The fuses in the running connections should be rated about 25 per cent. larger than the full-load current of the motor. This will allow the fuses to blow when the motor is overloaded over 25 per cent. for a long period, but as a fuse will carry 50 per cent. overload for several minutes there is no danger of the fuse opening the circuit from brief overloads which the motor could easily carry without harm.

The starting current of a squirrel-cage motor is several times the full-load running current, but owing to the fact that the starting period is of short duration and a fuse takes an appreciable time to reach the melting temperature, the starting fuses need not be as large as they would have to be if they blew the moment the overload occurred.

Where circuit-breakers are installed instead of one set of fuses it is preferable to use them in place of the running fuses rather than the starting fuses, in order that the time-element feature of the fuses may be utilized during the starting period. Ordinary circuit-breakers, such as are usually installed for motor protection, open instantly when the current for which they are set is exceeded, and this lack of time lag makes the necessary current setting too high for proper protection during the starting period, but is not a serious objection when running. A few manufacturers mount the running

FIG. 2. BELL MOUTHS ON FUSE BOX

fuses directly on the starter, but for the reasons given above the connections of these fuse blocks should be changed to the starting position and larger fuses installed for use during the starting period.

Another important advantage obtained by the use of a circuit-breaker in the running position is that if a careless operator should throw the starter to the running position before the motor is up to speed, that will not cause a delay such as would be the case were it necessary to replace the blown fuses; the circuit-breakers may be closed and a fresh start made before the motor has come to rest.

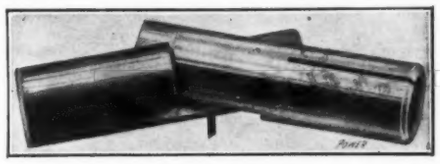


FIG. 3. A GOOD FORM OF CONNECTOR

Although the average employee outside of electrical workers regards a circuit-breaker with awe and will not tamper with it for any price, there are many cases where a machine operator will raise the setting of the breaker so that he will not be troubled to reclose it when opened by the careless handling of his starter. To prevent this practice, I have adopted the expedient of sealing the adjustment screws of all circuit-breakers with lead seals similar to those used on meter cases.

Protection of Live Parts

All live parts of a motor installation should be protected so that workmen cannot come into contact with them accidentally or carelessly. This applies equally to high- and low-voltage circuits, because it is not always the actual shock

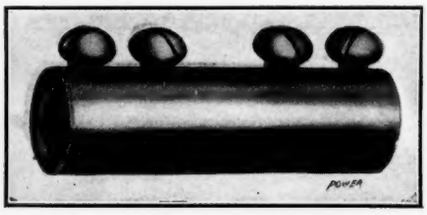


Fig. 4. A Poor Form of Connector

which causes injury in accidental contact with a circuit. The natural recoil which follows even a small shock is often the cause of a man's fall from an elevated platform or of his coming into contact with gears, belting, etc., which result in serious injury.

The switch should preferably be of the oil-break type. A convenient arrangement is an oil switch with automatic overload and no-voltage release magnets.

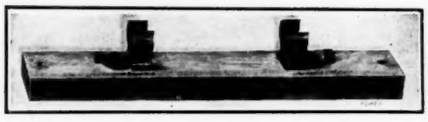


Fig. 5. Cartridge Fuse Holder

The fuses should be of the cartridge type inclosed in an iron box, as indicated in Figs. 1 and 2, with clamps by means of which the fuses may be withdrawn for inspection or renewal when the box is opened as shown in Fig. 1. This not only permits the safe renewal of fuses by any person, but also serves as an additional switch in case of trouble.

Where fuses and circuit-breakers are both installed, the fuses should always be on the line side of the circuit-breaker,

Fig. 6. Holder for Large Fuses

which will allow repairs to be made on the circuit-breaker when the fuses are out, and also allows the fuses to protect it in case of a flash-over due to lack of oil or other causes.

Where conduit work is installed, bell mouths, such as those shown in Fig. 2, should be used at all fuse boxes and circuit-breakers. This makes the conduit job continuous and protects the wires from mechanical injury. Where it is necessary to resort to open work, as in most types of auto-starters, a condulet with a porcelain cover having the required number of holes should be used to terminate the pipe.

The leads from auto-starters and motors should not be permanently soldered to the connecting wires. Knife-blade connectors of the type shown in Fig. 3 should be used instead; their extra expense will be more than offset in the time saved by their use whenever any of the apparatus requires change of connections or replacement while under repair. Instead of taping up the connectors, a better, neater and more accessible job will be made if a piece of fiber tubing, such as the casing of a blown fuse, be slipped over each connector and held in place by one or two turns of tape at each end. Connectors of the type shown in Fig. 4 are disapproved by the underwriters and should never be used on permanent work, and switches, fuse blocks, etc., having binding posts of the type where the wire is held in a drilled hole by the pressure

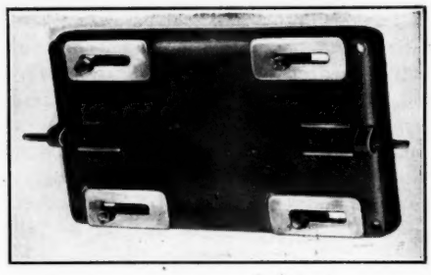


Fig. 7. Motor "Base" for Ceiling Mounting

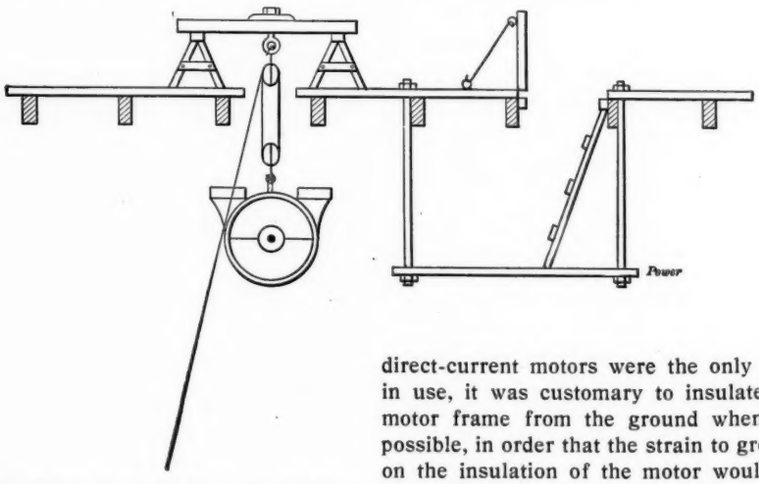


Fig. 9. Method of Placing Inverted Motor

of a screw point should be avoided. The contact secured is so small that the screw easily loosens under vibration and an arc starts between the wire and the binding post. Several cases have come under my observation where this arc has increased, unnoticed, until the heat of the wire ignited the insulation; the liability to cause a serious fire this way is obvious.

For currents not exceeding 30 amperes, the underwriters approve washer and screw connections of the type shown in Fig. 5, with spring jaws to take fuses having ferrule terminals. For currents exceeding 30 amperes, the soldered lugs of the type shown in Fig. 6 must be used at all switches and fuse blocks, and for currents exceeding 60 amperes knife-blade terminals to fit jaws of the type shown in this figure are required on all fuses.

A few years ago motors used to be equipped with exposed terminals, but now nearly all motors are furnished with flexible leads and connectors of the type shown in Fig. 3.

Fig. 8. Drip Pan for Ceiling Motor

This greatly reduces the liability of accidental short-circuits, and any strain on the terminals is prevented from injuring the interior connections of the motor by a clamp of insulating material applied to the flexible cables.

Location of Motors

When motors are mounted on machine tools for direct connection or gearing, or on structural-steel foundations attached to a steel-frame structure, the motor frame will, of course, be grounded. In the early days of the electric drive when direct-current motors were the only kind in use, it was customary to insulate the motor frame from the ground whenever possible, in order that the strain to ground on the insulation of the motor would be minimized and also to permit the temporary operation of a grounded motor. The superior quality of insulation used in modern motors makes the windings, etc., very unlikely to become grounded, and it has therefore become standard practice to ground the frames of all induction motors, a special ground wire being run when motors are mounted on a wooden foundation. Common humanity demands this in the case of high-voltage machines, but even on low voltages a shock may result in serious injury, as explained in the previous article.

Another reason for the grounding of induction-motor frames is to prevent the accumulation of static charges which occur on an insulated motor, due to its static capacity with respect to the ground. Varying-speed crane motors of the wound-rotor type appear to be especially liable to this trouble, and several cases have come under my observation, when these motors were mounted on a wooden-frame crane, where the crane could not be operated until the frame of the motor was grounded. The shocks only occurred occasionally, but the workmen would not handle the crane until the ground wire was installed. If a direct ground on the frame is not desired 500 to 1000 ohms of dead resistance may be connected in the ground wire, which will allow the static charges to escape, but the motor frame may have a dangerous potential above the earth, in case of an actual ground in the windings.

The location of a motor will naturally be governed by local conditions, but a clean, light and well ventilated place must be chosen, if repairs are to be kept down to the minimum. Lack of ventilation reduces the load the motor can carry and a dark, dirty or inaccessible location renders it liable to be neglected. When the saving in floor space renders it advisable to place the motor on a side wall, or a better arrangement of belting can be made by installing the machine on the ceiling, some means of ready access should be furnished or the motor is sure to be neglected. For ceiling installation a special bedplate is required, as shown in Fig. 7, with a drip pan (Fig. 8), held in place by an eyebolt, to prevent surplus oil from dropping to the floor or machines below.

The easiest way to install a ceiling motor is to tap a hole in the bottom of the frame, if one is not already there, and cut a small trap in the floor above, through which the tackle may be dropped to hoist the motor into place, as represented in Fig. 9. For very high ceilings another trap may be cut to one side, to give access to the motor for inspection, and a working platform hung under this trap, as in Fig. 9.

LETTERS

The Three-wire versus the Two-wire Generator

At several engineers' clubs during the past month, I have heard the subject discussed as to whether the three-wire direct-current generator with collector rings is better than the two-wire generator with a motor balancer. Personally, I believe the three-wire machine to be the better for the following reasons:

With the three-wire generator there is a saving in that only one machine has to be bought instead of two or three; lower maintenance, because there are fewer parts to be cared for; and higher efficiency, because one machine has smaller losses for a given output than two or three machines.

Some engineers argue that the fluctuations in voltage, due to a continually changing load, cannot be overcome without a motor balancer. I have five electric elevators running from a 250-kilowatt generator and drawing 1000 amperes. Sometimes the load of all the elevators is thrown on at once, but the voltage drop is not noticeable on either the lights or the engines. This shows that a three-wire generator can handle a sudden load just as well as a compensating set. At times I have seen a drop or rise in voltage, but this was through no fault of the generator; it was due to the governor not being sensitive enough to respond to the sudden change.

Furthermore, when the circuit breaker goes out, the place is in total darkness until the compensator is built up again. On a three-wire machine, the pilot lights remain lighted and it is only necessary to pull the main switch, throw in the circuit breaker, then close the main switch, and the load is on. I believe this to be the best and simplest way, but would welcome other opinions upon the subject.

WALTER N. SMITH.

Chicago, Ill.

Overheating of Copper Plated Dynamo Brushes

Concerning the burning of brush faces on generators, of which there has been some discussion lately, an experience which I once had with this kind of trouble may be of interest. The machine was a 150-kilowatt direct-current compound-wound generator, having six sets of brushes, four in each set, and delivering 250 volts at 200 revolutions per minute. When this machine was started some of the brushes became white hot in spots near the tips of the brushes. It was at first thought this was due to poor contact between the brushes and the commutator but examination proved the contact good. The burning would change from one brush to another until nearly all were destroyed in the center, which took about three days. Sometimes a brush would burn when carrying about one-third of the load and would not burn at full load. Different grades of brushes were tried but that did not help the trouble any. There was no sparking except at the brushes which were heated. The holders were of the box type and had the usual shunt or "pigtail" connections to the brushes.

While watching the machine one day it occurred to me that perhaps the vibration of the brushes in the holders might improve the contact between the holder and the copper plating of the brush and lower the resistance there so much that the brush would carry more than its share of the current. Scraping off the copper plating on the brushes proved this reasoning to be sound and corrected the trouble. The machine has been running with full load for three years since the copper plating was scraped off and the same brushes are in use yet. The pigtails are of ample size to carry the current without depending on contact between the brushes and the holders. They are clamped on and the plating was not removed under the clamp. I have sometimes tried a single copper-plated brush on this machine and it immediately becomes white hot at the contact end.

P. L. WERNER.

McKeesport, Penn.

An Unbalanced Air Gap and a Powerless Motor

A day or two ago I had an experience with a 35-horsepower three-phase induction motor which was new to me. The moter is in an out of the way department and an employee came running to the engine room to say that the motor had "gone up." I hurried to the spot and found that the foreman had shut off the current and the motor was at a standstill. I was told that the motor had made a noise like a circular saw and lost speed. I got up on the suspended platform on which the machine was mounted and found the bearing on the belt side so hot that the babbitt had started to run. Pulling off the belt, I turned the motor shaft by hand and found that it turned freely, and apparently without the rotor touching the stator at any point. I put the compensator handle at the "starting" position and the motor ran at about 15 revolutions per minute and hummed a great deal. Next I tested all the fuses and the compensator fingers and found them all right; tests for open circuits and grounds also failed to show anything wrong. But the motor simply would not run normally.

Then I took the rotor out and found that it had rubbed at one spot on the stator, but had done no harm. After the burned bearing had been babbitted and fitted to the shaft again, the motor was reassembled, the current turned on and away she went as smoothly as a top. Now the question is why would not the motor run at first? All I did that would affect its operation at all was to put the rotor in the center of the stator bore, equalizing the air gap all around. I wish someone would explain why lowering one end of the rotor until the air gap is nearly all on one side at that end apparently robbed the motor of all its power.

R. MANLY ORR.

Brantford, Ont.

Gas Power Department

Elementary Lectures on the Gas Producer

By Cecil P. Poole

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Peat

As stated briefly in the last lecture,* peat is merely decomposed vegetable matter, consisting chiefly of decayed moss and water plants. Most of the peat is found in bogs, those in fairly high altitudes containing decayed swamp moss and small forest plants while those in low altitudes, located near rivers and creeks, contain decayed grasses, leaves and water plants. As would be expected, fresh peat is so moist that it is useless for fuel; it must be dried out considerably before it will burn readily, and in order to be of commercial value as a fuel it has to be put through an elaborate process of drying and pressing.

Even when dried as far as practicable, peat contains a large proportion of moisture in comparison with lignite and coal. Two typical analyses of air-dried peat are given in Table 2 herewith. The "heat values" given at the foot of the table are the numbers of heat units liberated by the combustion of one pound of peat.

TABLE 2. TWO PEAT ANALYSES.

The figures are percentages, by weight, of air-dried peat.

	Per Cent. Bohemian.	Per Cent. Irish.
Carbon	39.26	53.18
Hydrogen (available)†	3.88	5.54
Nitrogen	1.36	1.91
Moisture	26.26	25.12
Ash	17.27	9.93
Unknown	11.97	4.32
Heat value	68.40	94.15

Dried peat uncompressed is very porous and ignites readily at about 375 degrees, Fahrenheit. It burns in the open air with a long, smoky flame. Because of the expense of drying and pressing it, and the high cost of transportation (due to its bulkiness), peat is not widely used for fuel.

Lignite

Lignite is commonly called "brown coal" because of the color of most lignites. There are, however, black lignites as well as brown. Table 3 gives representative derived analyses‡ of lignites, which show the wide range of moisture and ash contained in the fuel. The North Dakota sample, for instance, contained over 35 per cent. of moisture and 9.3 per cent. of ash, whereas the Wyoming peat contained only 9.4 per cent. of moisture and nearly 21 per cent. of ash.

*The fourth of the series, which appeared in the issue of August 16.

†All hydrogen except that contained in the moisture.

‡See page 1476, issue of August 16.

The "heat value," given at the bottom of each column of the table, is the number of heat units that are "liberated" or "evolved" by burning one pound of lignite.

The percentages in the body of the tables are the proportions, by weight, of the various substances in the lignite. For example, the Montana lignite showed 59.08 per cent. of carbon; that means that in each pound of the lignite there was 0.5908 pound of carbon, or in 100 pounds of the lignite there were 59.08 pounds of carbon. Some of this was solid carbon, called "fixed" carbon, as explained in the last lecture, and the remainder was gaseous carbon, combined with hydrogen to form hydrocarbon gases, but this feature of the carbon content need not be taken into consideration just now, beyond the fact that the hydrocarbon gases which contain the gaseous portion of the carbon also contain a portion of the available hydrogen.

TABLE 3. ANALYSES OF SOME REPRESENTATIVE LIGNITES.

The figures in the table are percentages, by weight, of the lignite as received.

Where found....	California.	Colorado.	Montana.	North Dakota.	Texas.	Wyoming.
Carbon	47.18	57.46	59.08	41.87	39.25	54.91
Hydrogen*	3.87	4.00	4.14	2.85	3.08	3.88
Sulphur	3.05	0.55	1.73	0.48	0.79	0.59
Nitrogen	0.66	1.15	1.33	0.69	0.72	1.02
Moisture	18.51	18.68	11.05	36.78	34.70	22.63
Ash	15.49	5.99	10.97	5.09	11.20	4.50
Unknown	11.24	12.17	11.70	12.24	10.26	12.47
Heat value	8,525	10,143	10,539	7,204	7,056	9,650

*Free hydrogen and that combined with gaseous carbon, forming hydrocarbons.

Bituminous Coals

From the name, the student would naturally suppose that bituminous coals contain bitumen, but they don't. In early times, when the scientific knowledge of geology in general, and coals in particular, was very imperfect, it was thought that certain soft coals contained bitumen because they felt resinous to the touch and burned very freely and with a long flame. Hence, they were called bituminous, and the name has clung to all soft coals above the grade of lignite.

There are two general classes of bituminous coals, known as "caking" and "noncaking." Caking coal, when thrown on a fire, swells up and the lumps fuse together, or "cake," in a sort of spongy mass which must be broken up frequently to allow the draft to get through and the fire to get at the particles of coal. Noncaking coal does not behave that way.

Table 4 gives the derived analyses and heat values of twelve bituminous and three semi-bituminous coals which are fairly representative of these grades. A glance at the proportions of carbon and hydrogen will show that the ratio of carbon to hydrogen is considerably higher in semi-bituminous than in bituminous coals. A still further increase in this ratio carries the classification up to anthracites.

Anthracites

Anthracite is a hard, lustrous coal, composed almost entirely of carbon. Semi-anthracite comes between anthracite and semi-bituminous coals. It is mostly lumps, as it comes from the dealer, but the lumps can be crumbled by a good sharp blow, whereas "straight" anthracite is almost as hard as rock and is not easily fractured or crushed. This distinction, of course, is meant to apply to grades not very near the technical dividing line between the two classes. A semi-anthracite which lacks only 1 or 2 per cent. of coming within the anthracite class cannot be distinguished by sight or touch from an anthracite that is just over the dividing line between the two classes.

Heat Value of Coal

It is impossible to compute the actual heat value of coal from any analysis, because of the imperfection of all known methods of analysis, but a reasonably close idea of the heat value may be obtained from the derived analysis by simple calculation, as follows:

Add the percentages of carbon, hydrogen and sulphur and multiply their sum by 170, *for lignites and soft coals. Deduct from this product* 10 *times the percentage of moisture.*

For semi-anthracite, use 160 *instead of* 170 *as the first multiplier.*

For anthracite, use 150 *instead of* 170 *as the first multiplier.*

These rules are based on the assumptions that the total combustibles in lignites and soft coals average about 17,000 heat units per pound, those in semi-anthracite about 16,000 and those in anthracite about 15,000; and that the moisture in all fuels absorbs about 1000 heat units per pound when it is evaporated into steam. As the proportions are expressed in percentages, the heat values must be divided by 100 in order that the product of the two shall be correct. That is, multiplying 80 per cent. by 170 gives the same result that would be obtained by multiplying 0.80 by 17,000; and if coal contains 80 per cent. of combustible, each pound of it will contain 0.80 pound of combustible.

Expressing the rules in the shape of formulas:

For lignite and soft coal—

$$170\,F - 10\,M = \text{B.t.u.}$$

For semi-anthracite—

$$160\,F - 10\,M = \text{B.t.u.}$$

For anthracite—

$$150\,F - 10\,M = \text{B.t.u.}$$

In these formulas F = the sum of carbon, hydrogen and sulphur percentages; M = percentage of moisture, and *B.t.u.* = heat units per pound of lignite or coal.

The chart, Fig. 13, gives the gross heat for different percentages of total combustibles. In using it, it is necessary to subtract 10 times the moisture percentage from the result obtained from the chart.

Examples

The Colorado lignite referred to in Table 3 contained

	Per Cent.
Carbon	57.46
Hydrogen	4.00
Sulphur	0.55
Combustible	62.01

The heat liberated by burning these combustibles is approximately

$$170 \times 62 = 10{,}550$$

B.t.u., and as the lignite contained 18.68 per cent. of moisture, the heat used up in evaporating this into steam would be about

$$10 \times 18.68 = \text{practically } 187$$

heat units. The estimated net heat of the lignite, therefore, is about

$$10\,550 - 187 = 10{,}363$$

B.t.u. per pound. The test, however, showed 10,143 (see Table 3), so that the estimated heat value is too high by

$$10{,}363 - 10{,}143 = 220 \text{ B.t.u., or}$$

$$\frac{220 \times 100}{10{,}143} = 2.17$$

per cent.

The Texas lignite contained

	Per Cent.
Carbon	39.25
Hydrogen	3.08
Sulphur	0.79
Combustible	43.12
Moisture	34.70

The gross heat is about

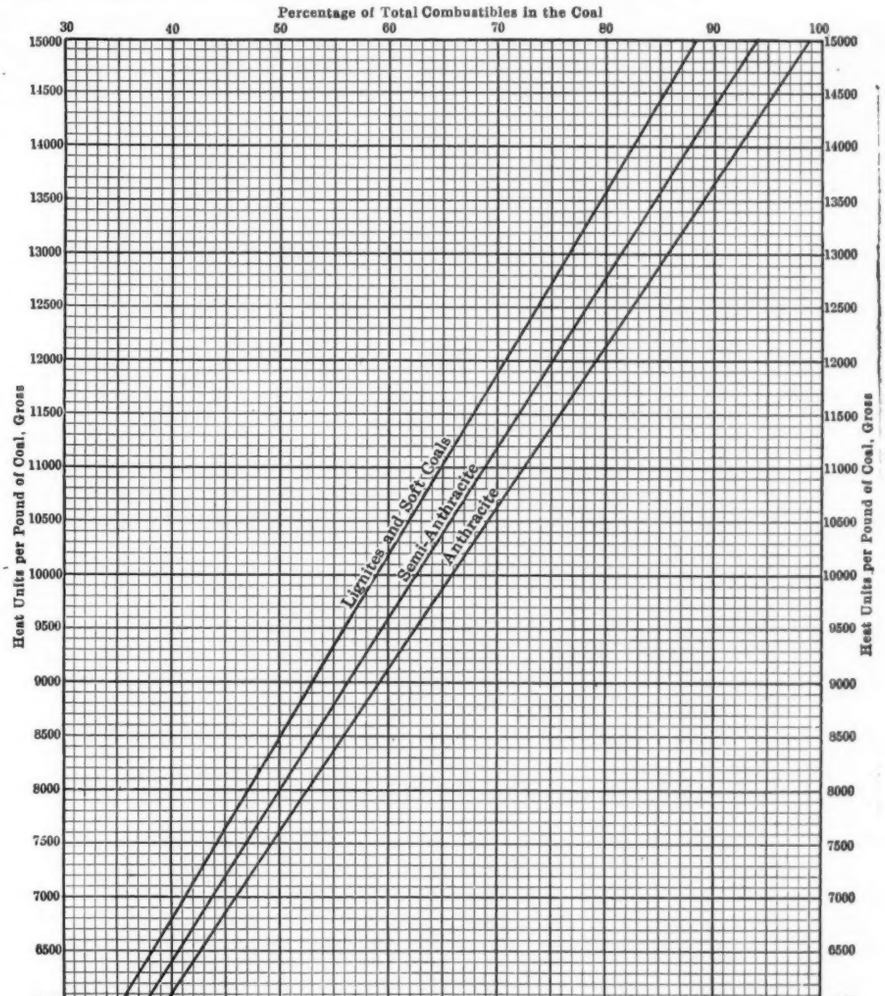

Percentage of Total Combustibles in the Coal
Heat Units per Pound of Coal, Gross
Lignites and Soft Coals
Semi-Anthracite
Anthracite
Percentage of Total Combustibles in the Coal
Heat Units per Pound of Coal, Gross
Power

FIG. 13. RELATION BETWEEN TOTAL COMBUSTIBLE AND GROSS HEAT IN COALS

$$170 \times 43.12 = 7330 \text{ B.t.u.}$$

The heat absorbed by evaporated moisture is about

$$10 \times 34.7 = 347 \text{ B.t.u.}$$

The estimated net heat in the lignite, therefore, is

$$7330 - 347 = 6983$$

B.t.u per pound. The heat value by test was 7056, so that the estimated value is $7056 - 6983 = 73$ heat units, or 1.03 per cent., too low.

The Illinois bituminous coal of Table 4 contained 60.74 per cent. carbon, 4 per cent. hydrogen, 1.32 per cent. sulphur and 12.91 per cent. moisture. Total combustible, 66.06 per cent.

$$170 \times 66.06 = 11{,}230 \text{ B.t.u.}$$

for the gross heat.

$$10 \times 12.91 = 129$$

B.t.u. lost in steam, and the estimated net heat is about

TABLE 4. ANALYSES OF REPRESENTATIVE SOFT COALS.

Bituminous.	Carbon.	Hydrogen.*	Sulphur.	Nitrogen.	Moisture.	Ash.	Unknown.	B.t.u.
Alabama	68.69	4.47	1.01	1.54	3.36	12.43	8.50	12,350
Arkansas	75.31	3.93	1.28	1.53	2.19	11.63	4.13	13,464
Illinois	60.74	4.00	1.32	1.15	12.91	11.64	8.24	10,804
Indiana	60.34	4.10	2.50	1.18	11.40	13.40	7.08	11,061
Indian Territory	62.34	4.24	3.96	1.36	6.24	13.21	9.45	11,228
Iowa	59.82	3.90	5.03	0.94	8.24	16.00	9.07	11,027
Kansas	61.88	4.23	6.27	0.92	4.18	17.91	4.61	11,642
Kentucky	65.29	4.51	3.52	1.40	7.92	10.06	7.30	12,022
Missouri	57.00	4.05	5.25	0.94	8.33	19.36	5.07	10,586
Ohio	72.03	2.93	2.19	1.07	8.25	6.72	4.81	13,591
Pennsylvania	76.15	5.26	2.52	1.56	1.52	6.42	6.57	14,762
West Virginia	78.00	5.09	0.90	1.54	1.75	6.34	6.38	14,107
Semi-bituminous.								
Arkansas	76.44	3.56	1.99	1.37	2.36	12.08	2.20	13,259
Pennsylvania	83.94	4.21	0.91	1.27	0.59	6.04	3.04	14,753
West Virginia	82.87	4.59	0.65	1.68	1.53	5.05	3.63	14,807

*Free hydrogen and that contained in hydrocarbons.

$$11{,}230 - 129 = 11{,}101$$

heat units per pound. The test showed 10,804, so that the calculated value is 2.75 per cent. high.

The Pennsylvania bituminous coal (Table 4) contained

	Per Cent.
Carbon	76.15
Hydrogen	5.26
Sulphur	2.52
Combustible	83.93
Moisture	1.52

The gross heat is about

$$170 \times 83.93 = 14{,}268 \text{ B.t.u.},$$

and subtracting the heat lost in the steam ($10 \times 1.52 = 15$) gives

$$14{,}268 - 15 = 14{,}253$$

heat units per pound, net, which is 3.45 per cent. less than the test figure of 14,762.

TABLE 5. ANALYSES OF REPRESENTATIVE HARD COALS.

	Pennsylvania Semi-Anthracite.	Colorado Anthracite.
Carbon	72.65	86.46
Hydrogen*	2.50	1.62
Sulphur	0.74	0.29
Nitrogen	0.77	0.75
Moisture	5.41	3.45
Ash	15.78	5.90
Unknown	2.15	1.53
B.t.u.	12,047	13,470

*Free hydrogen and that contained in hydrocarbons.

The West Virginia semi-bituminous coal contained

	Per Cent.
Carbon	82.87
Hydrogen	4.59
Sulphur	0.65
Combustible	88.11
Moisture	1.53

Gross heat of combustibles:

$$170 \times 88.11 = 14{,}979$$

heat units, and the steam absorbs about $10 \times 1.53 = 15$ heat units, leaving

$$14{,}979 - 15 = 14{,}964$$

heat units per pound, which is 1.06 per cent. more than the test showed (14,807).

A Pennsylvania semi-anthracite contained

	Per Cent.	
Carbon	72.65	
Hydrogen	2.50	
Sulphur	0.74	
Combustible	75.89	
Moisture	5.41	
Gross heat = 160 × 75.89 =	12,142	B.t.u.
Steam absorbs 10 × 5.41 =	54	B.t.u.
Net heat per pound	12,088	B.t.u.

The heat value obtained by test was 12,047 B.t.u. per pound.

A Colorado anthracite contained

	Per Cent.	
Carbon	86.46	
Hydrogen	1.62	
Sulphur	0.29	
Combustible	88.37	
Moisture	3.45	
Gross heat = 150 × 88.37 =	13,255	B.t.u.
Steam wastes 10 × 3.45 =	35	B.t.u.
Net heat, estimated	13,220	B.t.u.

The heat value obtained by test was 13,470 B.t.u. per pound.

The foregoing examples, of course, were figured in detail. The chart does not give readings as closely as the estimated values are computed, but it will serve very well to get at a rough approximation of the gross heat value, from which to subtract the heat used up in evaporating the moisture.

Some Points in the Practical Operation of Gasolene Engines

By A. L. Brennan, Jr.

The ignition system of any internal-combustion engine is a fruitful source of "trouble" and should be investigated whenever an engine stops or refuses to run without an evident cause. The wiring plans of different engines vary somewhat, but the general principle is the same in all of them, so I shall not go into this subject at all, but will assume that the engine is wired correctly.

The vibrator or "trembler" spring of an induction coil should be adjusted to the least possible tension consistent with reliable firing, and it will be found necessary to make slight changes from time to time, particularly when running on battery current, because as the battery voltage diminishes, the tension on the vibrator spring should be reduced. All contact points must be clean, flat and in line, to insure proper contact. Pitted points can be readily cleaned and "squared" by using a very fine flat file.

Before starting an engine the following details should be given careful attention. Fill all grease cups and lubricators. If the engine oils from the base, see that there is a sufficient quantity of clean oil in the base. Turn on the circulation water to the jacket, open the discharge outlet if necessary, turn on gasolene, close the switch in the ignition circuit and crank. If the engine refuses to start, any one of the following conditions may be the cause: A loose terminal or a disconnected wire, usually to or from the battery; damp cylinders; battery plug not in position; a carbonized or broken spark plug; dirt in the gasolene pipe; mixture faulty; water in the carbureter; the check valve in the carbureter not working; inlet valve stuck; poor compression; leak in the water jacket, allowing water to get in the cylinder; engine flooded with gasolene; battery run down; timer slipped on the shaft; muffler or exhaust pipe clogged.

This is a rather formidable list of troubles, but only a few of them occur in every-day operation. The best thing to do first is to look over the electrical system for a broken connection, and a good way to test for this is to close the switch, turn the engine over slowly and note if the coil vibrator buzzes (assuming that the jump-spark system is used). If the vibrator works, then the trouble cannot be in any part of the primary or low-tension circuit, but there may be trouble in the high-tension wiring, spark plug, or the secondary winding of the induction coil. To test the plug it will have to be removed from the cylinder and laid on the cylinder, with the wire attached to it as before; then turn the engine over and note if there is a good spark at the plug points when the vibrator buzzes. If the plug shows a large even spark of a bluish tinge, the trouble is not in the electrical system.

On the other hand, if the vibrator on the induction coil does not buzz when the engine is turned over, one or more of the following troubles exists: A broken or disconnected wire between the coil and the engine; a short-circuit or a broken wire in the battery or generator circuit; contact points out of adjustment or in bad condition, failing to make good contact; binding posts on the coil broken or loose; the coil is wet or otherwise damaged; vibrator spring broken or vibrator stuck; battery weak or exhausted; wiring put back wrong. If the coil seems to work all right but the plug fails to fire, the seat of trouble is either in the high-tention ground wire or in the plug itself. If it be in the plug, it may be due to the parts being loose; the porcelain being broken, carbonized, or short-circuited by dampness.

Irregular explosions in the cylinder are usually caused by a weak battery, but may be due to broken-down insulation on wires, causing a partial short-circuit; a dirty or broken spark plug; wire terminals not being securely fastened; too much lubricating oil, or an inferior quality which carbonizes readily; vibrator points on the coils being out of adjustment or pitted; contacts in the timer being worn; carbureter check valve not working right; carbureter float not moving freely; water in the carbureter; gasolene feed pipe being clogged.

When an engine fires regularly but does not develop its normal power, the cause may be insufficient lubrication of either the bearings or the piston; the crank shaft being out of line; a poor grade of gasolene; wrong proportioning of the mixture; low compression due to stuck or worn rings, wear of the cylinder wall, a loose spark plug or loose valves or valve cages; insufficient lift of the exhaust valve; exhaust port partly clogged; exhaust cam out of place; too much tension on the air valve of the carbureter; inlet-valve spring out of order.

Muffler explosions are due to misfiring, the unburned mixture being delivered into the hot exhaust passages where it is ignited; misfiring may be caused by the exhaust valve being hung up or the stem being bent and preventing the valve from seating properly; by the mixture being too weak, or by an insufficient or retarded spark.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Dimensioning Pipe Bends

When the average engineer desires to get some pipe bends made, he naturally endeavors to make a sketch of what he requires. Many times, especially when the work requires anything but a 90-degree bend, the measurements given on the sketch submitted to the pipefitters are not definite enough. It is absolutely necessary that the angle between the straight runs of pipe be given in some manner, and that the ends of the pipe be definitely located.

It is quite common to see a sketch like Fig. 1 with the dimensions *A* and *B* given only. Now, if it is tried to make a full-sized drawing on the floor from these

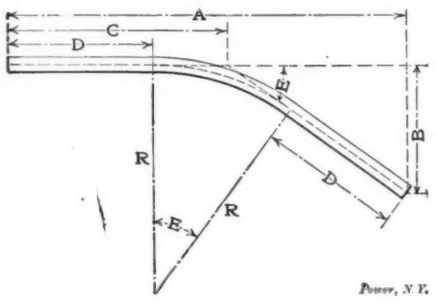


FIG. 1

two dimensions, it will soon be found that the bend may occur at several different points. By the addition of dimension *C* from one end of the pipe to the intersection of the center lines projected, the bend is definitely located and the direction of the pipe determined.

It is often thought that if *A*, *B* and the radii *R* are given, these are all the dimensions necessary, but this is not so. Di-

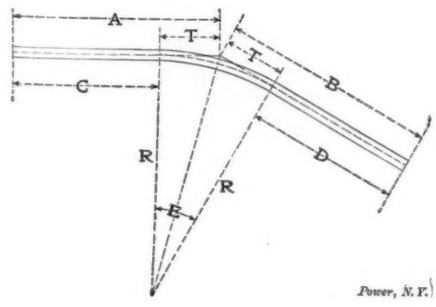


FIG. 2

mensions *C* or *D* will complete the requirements. Unless all the dimensions are taken with care, there is liable to be some difficulty in making a sketch to satisfy all these conditions. Dimensions *A*, *B* and *C* are the best to work from, with the radius *R* if the job is particular. If the angle *E* can be measured with a protractor, then the dimensions from the intersection of the straight runs to the ends of the pipe on both legs are the best ones to give.

The pipefitter must know at what point to start to bend the pipe, on what radius to bend it, how much pipe to bend, and the angle that the straight ends make with each other. His final check must be the angle and the length from the intersection of the center lines to the end of the pipe on each leg. This dimension is called the center-to-end length.

Suppose, as shown in Fig. 2, there are given the center-to-end distances *A* and *B*, the angle *E* and the radius *R*. In order to get the length of the straight ends *C* and *D*, *T* must be calculated and subtracted from *A* and *B*. Connecting the center point with the center of the arc of the bend will give two equal right-angled triangles.

The ratio found by dividing the perpendicular by the base of a right-angled triangle is called the tangent of the angle opposite the perpendicular. Most handbooks of engineering have tables which show this ratio corresponding to different angles. The angle to be considered is one-half of the angle *E*. Looking in a table of tangents the decimal number corresponding may be found, and as before stated this will be equal to *T* divided by *R*; or in other words, *T* is equal to *R* times the decimal found in the table. Subtracting *T* from *A* and *B* will give the straight lengths *C* and *D*.

The length of pipe to be bent must now be determined. It is known that the diameter of a circle times 3.1416 gives the circumference, and since the radius is one-half the diameter, the radius times 6.2832 equals the circumference, or the length of an arc of 360 degrees. Therefore the radius times 6.2832 divided by 360, or 0.01745, gives the length of an arc of one degree and unit radius. To get the length of any arc of a given angle and radius, multiply the radius by 0.01745 and by the number of degrees in the angle. By adding together dimensions *C* and *D* and the length of the arc, or bent part of the pipe, the amount of pipe to be cut will be obtained.

WILLIS M. WRIGHT.

New Haven, Conn.

Sand Erosion

Some years ago, on investigating a case of so called electrolysis, I found a condition of affairs as shown in the accompanying photograph, and traced the cause to sand erosion or sand blast, due to the rapid flow of water and sand against the pipe.

Since then, I have had occasion to notice this same occurrence in a number

EFFECTS OF SAND EROSION

of instances on water-works piping, and also to a more limited extent when any material leak occurred on steam piping laid in sandy soil. The results were far more rapid when water was present than when only steam escaped.

My conclusions are that it is not wise to lay steam pipes in the earth where a leaky connection is possible, and that the very best of workmanship is necessary for the installation of both steam and water pipes where laid in sand or gritty earth.

HENRY D. JACKSON.

Boston, Mass.

Water Supply Control

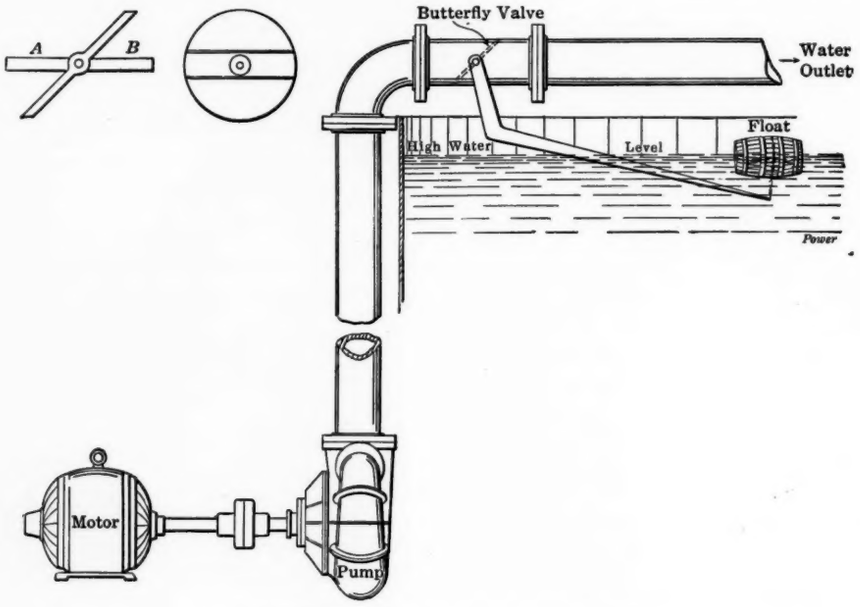
Frequently in power houses where a continuous water supply is needed, the rate of consumption varies considerably. With such conditions there is a risk of having at times an empty or an overflowing tank, or else it is necessary to have a man in the pump house to watch the water level.

If conditions will allow a centrifugal pump being used, the scheme shown in the accompanying illustration will solve the difficulty. The float is made of a common oak keg painted to prevent cracking, and the butterfly valve is one that any machinist can turn up on a lathe.

The method of turning a butterfly valve may not be familiar to all, hence a short

description may be of service. The casting or plate must have false stubs as *A* and *B*, which, when centered and placed on the lathe, allow it to be turned to the required diameter. These ends are then chipped off. The valve should be a fairly tight fit but not absolutely water tight as a small leakage will not matter. The valve may be fastened to the shaft by set screws.

A butterfly valve has as much pressure above the center as below, hence it may be termed a balanced valve. Thus no undue pressure is thrown on the lever connecting to the float. This is true in whatever position it is placed.



FLOAT ATTACHED TO DISCHARGE PIPE

Supposing the supply pipe to the tank, or, in other words, the discharge pipe from the centrifugal pump, to extend to an indefinite hight, the pump would pump as high as its capacity would allow and there the water would remain. This is practically the same as when a valve is placed at the end of a short discharge pipe, and no harm will result.

The operation of the scheme is obvious and I hope some brother engineer may profit by it.

FRANK S. BUNKER.

Vallejo, Cal.

An Emergency Decision

Some years ago I took a position with a new steel mill which was under construction and began by fitting up the first battery of five vertical water-tube boilers. Before it was possible to get a feed pump in commission I was called upon for steam with which to start the blowers on a battery of gas producers. There were a couple of new locomotives at hand, which had not been put into service, so an injector was taken from one of them and connected to the feed pipe at the rear of one boiler.

When the boiler was filled and fired I noted at what pressure the injector would work, finding that it refused to handle any water until the steam had reached 110 pounds. The boiler was rated at 300 to 500 horsepower and the size of the injector was only 1¼ inches. But as the full rating of the boiler was not required, it was possible, by leaving the injector on all the time, to keep sufficient water in the boiler. A night and a day fireman were put on and for several days things went all right. The gas producers were put into service and the open-hearth furnace started; they had reached the point where to cut off the gas was out of the question.

As the feeding apparatus for the other boilers could not be put into commission for some time, I began to get anxious, and cautioned the firemen to maintain the pressure so as to keep the injector going at all hazards. I lived about a mile from the plant, and was awakened a few nights after this by hearing the low-water alarm at the plant. Getting into a few clothes, I started for the plant at a full run, meeting the fireman, who was on his way to notify me. Arriving at the plant I found the injector stopped with the pressure gage on the boiler showing 80 pounds. Both furnace doors were wide open, but were choked with live coal and clinkers, which the fireman had attempted to withdraw. There was not a drop of water in the gage glass. I thought of the damage that would result from stopping the blowers and determined to raise the steam pressure in order to get the injector into operation. I accordingly pushed the fire back onto the grates, opened the dampers and began nursing the fire. In a few minutes it was white hot and the steam pressure began to rise, though at the time it seemed an age. In the meantime, I cooled the injector and when there was sufficient pressure it fairly shrieked as it caught the water. The boiler was kept going for a couple of weeks before another was ready to relieve it, but a thorough examination failed to locate a single injury. It is not known how low the water was in the boiler, but the cause of the trouble was undoubtedly due to the fireman having gone to sleep. He disappeared just after notifying me of the trouble and I have never seen him since.

T. J. BLOSS.

Chattanooga, Tenn.

Improving a Brake

The brake on our hoisting engine, located at the shaft of a mine, had always given more or less trouble. On the down trip it took strenuous work on the part of the engineer to set the brakes tight and quick enough to keep the cage under control.

Fig. 1 shows the kind of brake used.

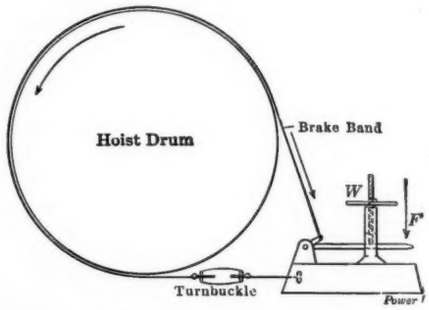


FIG. 1. ORIGINAL BAND CONNECTION

It was of the common band type, lined with hardwood blocks and tightened by means of either the handwheel *W* or the foot-lever *F*. The arrow denotes the direction of rotation of the drum when the cage was being lowered, and it will be seen that the pull on the brake strap in tightening was in a direction opposite to that of the drum. This made it very difficult to set the brake.

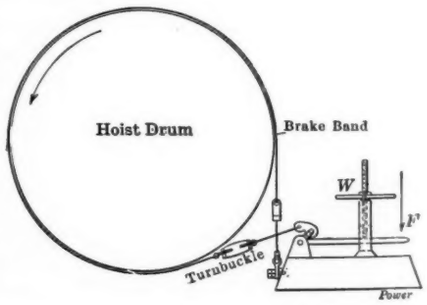


FIG. 2. BAND CONNECTIONS REVERSED

It was decided to overcome this trouble by reversing the connections of the brake strap, so that the pull in setting the brake would be in the same direction as the rotation of the drum. This was done as shown in Fig. 2, and proved

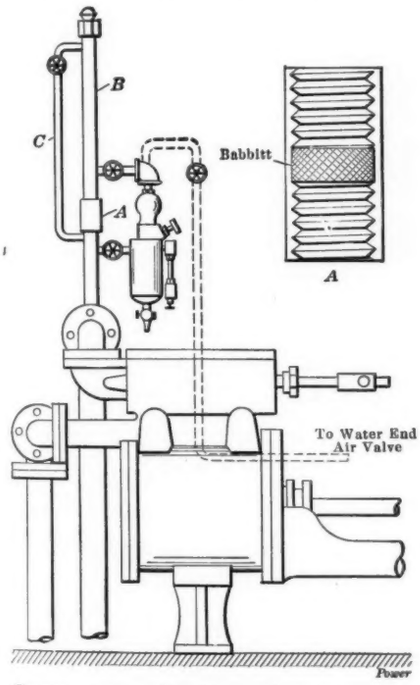
so satisfactory that a slight pressure of the foot on the lever *F* was as effective as screwing down on the handwheel formerly was.

J. A. CARRUTHERS.

Bankhead, Canada.

Piping a Lubricator

I was once asked to help doctor a boiler feed-pump lubricator, located on a standpipe rising from a low steam pipe, as shown in the sketch. It appeared that when the condensed-steam pipe was connected in the usual manner, the lubricator could not be made to supply the desired amount of oil. At the time I visited the plant, this pipe was connected to the water end of the pump, as shown by the dotted lines. Naturally the excessive water pressure soon emptied the lubricator, no matter how little the regulating valve was opened. The remedy applied was that of cutting a 12-inch length from the 1-inch standpipe, and inserting the galvanized coupling *A*. The middle portion of the coupling was filled with melted babbitt, which was made to adhere firmly to the sides. Then the remaining piece *B*, 18 inches in length, was screwed in place. The two chambers formed by the babbitted coupling were connected together by the ⅛-inch bypass *C*, and the lubricator piped as shown, the top connection being made above the coupling as near to it as practicable and the other connected below it. The water of condensation in pipe *B* is at all times sufficient to maintain a uniform flow of oil to the feed pumps, and the lubricator has given entire satisfaction since making the change.



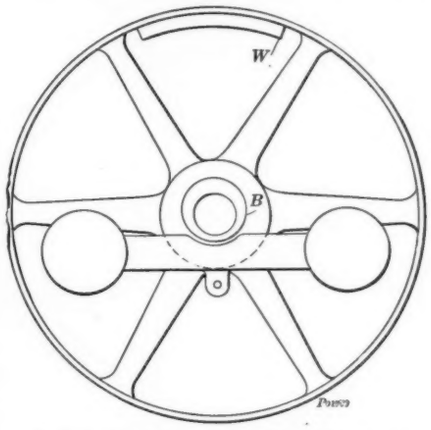
ORIGINAL AND IMPROVED ARRANGEMENT OF PIPING

H. P. EVERETTE.

Boston, Mass.

Flywheel Out of Balance

The engine was of the high-speed type and caused considerable annoyance by excessive vibration. At slow speeds it would run smoothly but upon speeding up to its normal speed the flywheel began to run out of true and the shaft began to spring. We placed a scribe to the hub of the flywheel and it traced the path *B*, shown in the sketch. This led us to believe that the flywheel was out of balance. We accordingly removed part of the weight *W* and experienced no more trouble.



FLYWHEEL SHOWING MARK MADE BY SCRIBE

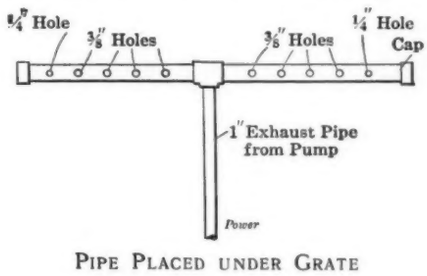
C. R. MCGAHEY.

Sheffield, Ala.

Preventing the Formation of Clinkers

Upon first coming to the plant at which I am at present employed, I was troubled with large clinkers forming on the grates, especially next to the bridgewall and along the sides of the furnace. I was about to run a jet of live steam under the grates when the thought came to me of using the exhaust from the pump, instead of allowing it to escape through the roof.

I accordingly drilled a hole through the front and center of the furnace, close up under the grates, also drilling several holes in two pieces of gas pipe long enough to reach underneath the grates. To give best results this pipe should be at least ¼ inch larger than the exhaust from the pump and the holes next to the ends should be smaller than the others, as shown by the accompanying sketch.



PIPE PLACED UNDER GRATE

Since installing this I have not been troubled with large clinkers and there is condensation enough in the steam on its way to the discharge pipe to dampen the ashes, so that when the ashes are pulled out there is no dust flying about.

HORACE B. WEST.

Geneva, O.

Erecting a Boiler

Some months ago our company purchased a 172-horsepower Babcock & Wilcox boiler located in an abandoned mill about three-quarters of a mile from our power house. The work of removing and reërecting this boiler devolved upon the writer, and a description of how it was accomplished may help someone having a similar task to perform.

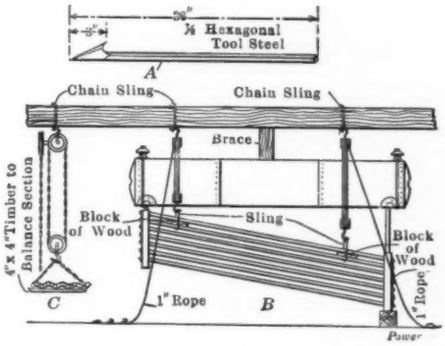


FIG. 1. METHOD OF REMOVING SECTIONS

With six negro laborers, a wagon, chainfalls, ropes, hammers and other necessary tools, I proceeded to the mill. After starting several men to work tearing out the brickwork, cautioning them to save as many whole bricks as possible, for use again in the new setting, with the others I commenced taking down the safety valves, steam pipe and doors, loading them onto the wagon as fast as they were removed to be hauled to the power house.

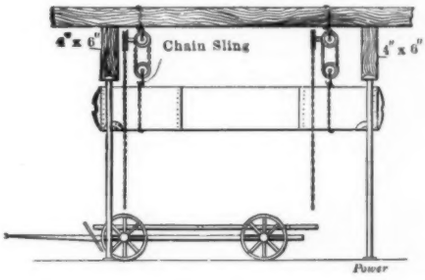


FIG. 2. LOWERING STEAM DRUM

This did away with frequent handling and at the same time kept the space around the boiler clear, allowing the men to work to better advantage.

At the power house the bricks were dumped on the ground, and a negro boy, at 75 cents per 1000, cleaned and stacked them. The iron work was brushed off

with a steel brush and given a coat of graphite paint.

When the brickwork was torn down to within 2 feet of the foundation, we braced the boiler from both sides with 2x4-inch scantlings. After all the brickwork had been removed, the nipples holding the mud drum were cut out, the mud drum was removed and blocking placed under the rear end of the headers, to steady them while cutting out the

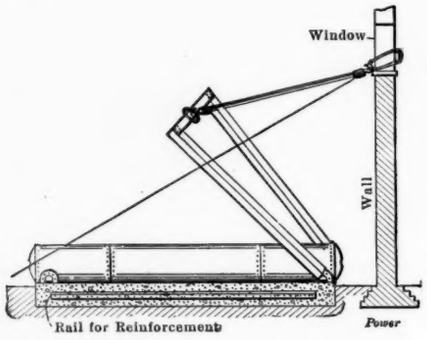


FIG. 3. RAISING THE COLUMNS

vertical risers, to allow for removing the sections.

At *A*, Fig. 1, is shown the form of tool used in splitting the nipples, care being taken not to cut the header or drum. At *B* is shown the manner in which the sections were removed, one section at a time; the slings being slipped along the tubes until the latter could be swung out to where the chainblock could be attached. Then the section was turned halfway around, as at *C*, ready to be lowered into the wagon.

Fig. 2 shows how the steam drums

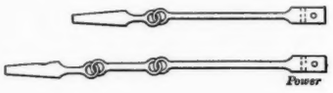


FIG. 4. SPECIAL EXPANDING TOOLS

were lowered onto the wagon after all the sections had been removed.

The drums were hauled to within a few feet of the foundation which previously had been put in and were set in position ready to be hoisted when the supporting columns were up. Fig. 3 shows how these columns and girders were raised. In setting the columns care was exercised to have them perfectly level and in line with one another.

The drums were hoisted into place in a manner similar to that in which they were taken down, and the cross pipes were bolted on to help hold them in place.

The tube sections were now raised and held by chainfalls while the new nipples were expanded into place. In expanding these nipples they were given an extra flare to insure against pulling out. Fig. 4 shows the special form of expanding pins used to get up through the handholes into the cast-iron headers.

As each section of tubes was placed in position the baffle bricks were fastened in place with a stout piece of string, the cast-iron pieces, which hold them from slipping, being put in and bolted together after all the sections were in place.

Next, the mud drum was blocked in place and the nipples expanded into it; after which the front was put on, the boiler closed and subjected to a hydrostatic pressure of 240 pounds per square inch, the boiler-feed pump being used for this purpose. With the exception of a few leaks around the handhole plates, the boiler withstood the test all right. After the setting was completed we kept a slow fire under it for several days to dry out the brickwork.

W. F. Cox.

Natchez, Miss.

Clogged Vent Pipe

An uptodate plant had been installed in a new office building, the dedication of which was marked by a large banquet in the evening. Consequently all the lights and fans were in use, which threw a heavy load on the power plant. The equipment included five Westinghouse engines of various sizes, four of which were in service on this particular occasion.

While the banquet was in progress and everything was apparently running smoothly in the engine room, there suddenly arose the smell of burning oil. Immediate investigation showed that the crank pins of two of the engines were running hot, and a speedy change was made to the spare engine. This had barely been accomplished when the other two engines began to smoke at the cranks. For a while it looked as if a shutdown were inevitable, for every time an engine was started the crank pins began to heat. Water was poured into the crank cases by the bucketful, but would not remain there.

In this type of engine the crank case has a vapor vent which must be kept open, or the accumulation of vapor will create sufficient pressure to force the water out through the overflow. Upon the water in the lower part of the case floats the oil into which the cranks splash at each revolution. In this case the vents from the crank cases of each of the five engines connected with the main vent pipe which led to the flue. Here was the cause of all the trouble—the opening into the flue was found to be stopped up. This caused excessive pressure in each crank case and forced the water out, thus placing the level of the oil out of reach of the crank.

EDWARD T. BINNS.

Philadelphia, Penn.

Clogged Suction Pipe

On taking charge of a 400-horsepower plant some years ago I was confronted with a complicated system of piping, the details of which it is not necessary to go into. The receiver tank rested horizontally on the floor. The two feed pumps rested on a foundation raised 8 inches above the floor beside the tank. The 2½-inch suction pipe was tapped into the bottom of the tank near one end, dropping about 3 inches to an elbow, and running 24 inches beyond the end of the tank to another elbow, finally terminating in a tee between the pumps. Just below the tank and past the ell was located a 2½-inch globe valve. The tank had not been opened for cleaning for over a year and the only opening to the atmosphere was through a ¾-inch vent pipe and a 1½-inch pipe with a stop cock was used for putting soda into the feed water.

One day the pump began to kick and the water began to lower in the boilers. The gage glass on the tank showed the water level to be near the top. The second pump was started but acted the same as the first. I concluded that the trouble was in the valve, and managing to supply the boilers with cold water from the small pump in the engine room, I took off the bonnet of the globe valve and allowed the water to drain from the tank. I saw at a glance where the trouble was and taking a packing hook picked out a handful of waste from the globe valve and pipe. How it came in the valve is still a mystery to me, as it was good strong waste showing very little sign of decay. It evidently had been put into the tank through the 1½-inch pipe.

R. A. CULTRA.

Boston, Mass.

Frost in the Engine Room

I am troubled a great deal during the winter months with frost accumulating upon the walls and ceiling of the engine room. The building is a frame structure, the outer wall of which consists of one ply of boards, one ply of sheathing and a covering of galvanized iron; the inside is sealed with ordinary lumber. The frost collects on the ceiling and is apt to fall on the generators and engine and cause trouble.

We have arrived at the supposition that the strong draft under the boilers displaces the air in the engine and boiler rooms, causing a suction of cold air through all the crevices in the walls.

We thought it might be possible to overcome the trouble by installing a cold-air duct from the outside to the ash pit. Opinions as to the advisability of this will be welcomed.

CHARLES FENWICK.

Saskaskia, Can.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Where Was the Westinghouse Governor?

In the June 14 issue Mr. Westerfield mentions the case of a runaway engine which was equipped with a Westinghouse governor. I should like to ask Mr. Westerfield where this plant was located.

Terrell, Texas. W. C. DRAKE.

Installing Globe Valves

In the June 21 issue, under the above heading, I mentioned an experience with globe valves on the bottom blowoff pipe of two boilers. The valves were of the angle type, with the pressure on top of the disk. The stem broke on one valve, leaving it closed, and on the other valve the thread on the stem became stripped and left this also closed. This plant was not located in Massachusetts.

Exceptions were taken to this part of my contribution by Mr. Smith in the July 26 issue. The threads on all valve stems wear out, but on none more quickly than where the pressure is on the top or back of the disk. In the case in question, I secured a 6-foot crowbar and a piece of board and put the block on the wheel. I then drove tne end of the bar into the brickwork and pressed the valve shut in less than two minutes in one case, fastening the free end of the bar with a rope and weight until the valve could be repaired. In another case I happened to have some long ⅜-inch bolts. Putting the heads in a vise, I bent hooks on them to go around the valves. Slipping the points through the wheel on the stem and placing washers on each bolt, I screwed up the nuts and had the valve clamped shut in a short time. But I have not seen the engineer who could open one of these valves which had the pressure on the top of the disk when the disk broke from the stem. I have used eight different types of stop cocks and fifteen different types of valves under both high and low pressures and have not found any which were perfectly satisfactory on a blowoff pipe under all conditions. A valve that is suitable for use under 200 pounds pressure with superheated steam is too small and too heavy for a pressure of 65 to 85 pounds. I have used the babbitt-faced disk mentioned by Mr. Smith and in two instances came to grief.

I wish to call Mr. Smith's attention to the acts of 1909 relative to the operation and inspection of steam boilers in Massachusetts. The section dealing with blowoff pipe and fittings states "Globe valves shall not be used." Referring to feed piping, it states that "When a globe valve is used on a feed pipe, the inlet shall be under the disk of the valve." These laws apply to boilers installed after May 1, 1908.

Perhaps Mr. Smith uses an iron-body blowoff valve having a heavy piston or disk about 2½ inches long, with a babbitt face at the bottom. With the long disk practically closing the port, and the babbitt face pressed against the seat, the valve would be practically steam and water tight. The only pressure at the seat, other than that caused by the screw in closing the valve, is from leakage past the disk. Therefore, the full boiler pressure does not come on the valve seat, but strikes the disk at right angles, leaving no more boiler pressure on the top of the disk than there would be on the bottom. Hence the valve can be opened very easily. This form of blowoff valve works very satisfactorily under pressures of 100 pounds or less, but cannot be classed as a globe valve on account of the shape of the disk.

R. A. CULTRA.

Boston, Mass.

I do not advocate valves set with the pressure on top of the disk. A short time ago I was called upon to open a 10-inch valve having the pressure above the disk, and even with the help of a 24-inch wrench applied to the wheel, I could not budge the valve. Of course, I could have used a little more force, but probably would have pulled the stem out of the disk.

The angle valves on the blowoffs of one of our boilers are also set in this way, and they are continually leaking at the glands, notwithstanding the fact that they are packed every year.

W. H. PLOWMAN.

Philadelphia, Penn.

Long Run of Pumping Machinery

Answering the request of Edwin D. Drefus in a recent issue of POWER, I would say that we have a 21 and 42x24.5 x36-inch compound-condensing Gaskell crank and flywheel pumping engine, rated at 7,500,000 gallons per twenty-four hours. This engine ran from June 1, 1901, to July 10, 1902, a total of thirteen months and ten days without a stop. This long run was necessary because all the other equipment was being removed, and a new 10,000,000-gallon pump was being installed. The Gaskell pumping engine was installed in 1873, and had been in service nineteen years when this record was made. It was thoroughly overhauled before starting in 1901 and never gave any trouble during this long endurance test; in fact, when it was shut down it was found to be in perfect working order. During this period an average of 5,000,000 gallons was pumped every twenty-four hours. The Holley system is used at this plant, and the pressures are 60 pounds for domestic purposes and 100 pounds on the fire lines.

JOHN ACKERMAN.

Auburn, N. Y.

Rim Bolts

In the June 5 issue, Mr. Mulham inquires whether the rim bolts in a wooden flywheel are in tension or in shear. A flywheel such as described and traveling at its customary speed would have, due to centrifugal action, a force tending to pull out the arms and the rim. If the rim were perfectly rigid there would be no forces acting on the bolts at all. But as the rim is actually more flexible than the arms, centrifugal force will cause the rim to assume an elliptical form due to its being held at two opposite points and a tension will be brought upon the rim bolts by the tendency of the rim to assume a circular form.

JOHN FRENCH.

Washington, D. C.

In reply to Mr. Mulham's letter in the issue of July 19, I offer the following: Both, the men who say the bolts will be in shear, and those who say they will be in tension, are right. The force acting in a direction along the radius of the wheel will equal $\frac{MV^2}{R}$ where M is the mass, i.e., the weight divided by 32.16, the acceleration due to gravity, V is the velocity of the rim, and R the radius of the rim. This is the force which places the bolts in tension and is present while the wheel is revolving. The shear upon the bolts varies directly with the acceleration of the wheel if running without a belt and if running with a belt there is the additional tension of the tight side of the belt. Hence, when the wheel is getting up to speed, even if there is no belt, there will be a tension on the bolts varying as the square of the speed and

there will be a shear upon the bolts which will remain nearly constant if the increase in velocity of the wheel is constant. When the wheel is running at constant speed with no belt the shear will be negligible except for a small amount due to the windage of the rim. With a belt, the shear will depend upon the tightness of the belt and the load placed upon it. The belt will hold the rim against the arms and the only stress upon the bolts under the belt will be the shear.

P. ARTHUR TANNER.

St. Louis, Mo.

The question as to the nature of the stress imposed upon the bolts holding the rim of a built-up flywheel to the arms, has been raised by Mr. Mulham in the July 19 issue. If the nuts were drawn up very tight on the bolts, a tensile stress would be created. If the arms were pocketed a shearing stress could not under any circumstances exist; a shearing stress is possible only where the arms are not pocketed and when the bolts are not strong enough to make the adhesion between the contact surfaces great enough to resist the tendency of the rim to slip. Possibly the fact that the belt pulls on the rim may have had some influence in creating the idea of a shearing stress, but in a properly designed wheel the strength of the various parts is such that the effect of the belt pull is negligible. Inasmuch as there is no shearing stress imposed on the bolts in a properly designed flywheel rim having pockets or recesses for the arms to fit into, there is but one type of stress which the bolts must resist, namely, tensional. Naturally, the centrifugal force of the rotating rim increases this stress on the bolts.

WESLEY MCARDELL.

Brooklyn, N. Y.

Induced Draft Trouble

Referring to Mr. Faulthier's letter in the July 12 issue, concerning trouble with an induced-draft system, I believe that his fan has a greater capacity than the smokestack, consequently causing a reaction in the current. If he were to reduce the speed of the engine he would obtain better results under the circumstances, as there is a limit to all things.

J. S. TURNER.

Big Rapids, Mich.

Replying to Mr. Faulthier's letter in the July 12 issue on "Induced Draft Trouble," I would say that, in my opinion, the trouble could be remedied by putting an elbow on the discharge pipe so as to discharge upward in the smokestack; this would act somewhat as an ejector. The probable cause for the blower not working under present conditions is that it discharges downward and the smoke just rolls around in the base of the stack, the blast from the blower being strong enough to act as a wall and prevent the smoke from passing up the stack. It should be remembered that a blower will not keep discharging into an air-tight receptacle.

FRANK GARTMANN.

Sheboygan, Wis.

Runaway Engines

In the July 19 issue, Mr. Trube cites the case of a runaway engine and tells how it was brought under control again by closing a stop valve in the steam line. His remarks show that the engine was not properly adjusted or it would not have run away at all under the conditions.

An engine should be run with the throttle valve wide open all the time. It is the duty of the governor to regulate the speed of the engine, regardless of the variations in pressure or load, and the throttle valve should never be used for this purpose.

Mr. Trube states that when the steam pressure was low the throttle valve had to be opened wide in order to keep the engine running up to speed, and as the steam pressure was raised the speed increased so that the throttle valve had to be closed somewhat; but as the throttle valve refused to work, the speed continued to increase until the engineer closed the stop valve, the regulation for the balance of the week being done by the stop valve. The engine was a cross-compound Corliss. I wonder where the governor was, or if it was regarded as only an ornament and had been discarded? It is quite plain that the engine was not correctly adjusted, the governor apparently being of no use whatever.

From the old type of plain slide-valve engine, governed by a throttling governor, to any of the modern types of engines, all should be operated with the throttle valve wide open. If an old slide-valve engine does run away, because of the governor belt having broken or jumped off, the quickest way to stop the engine is to grasp the governor balls and raise them, thus instantly closing the governor valve; this can be done quicker than any throttle valve can be closed.

In a single-eccentric Corliss valve gear the latest point of cutoff is 45 per cent. of the stroke and the earliest is zero, with all variations in the cutoff, from zero to maximum, under the full control of the governor. When the governor fails to work, the safety cams are brought into service, preventing the steam valves from being opened, and the engine stops. It is a very unusual defect that can cause a properly adjusted Corliss engine to run away.

Had Mr. Trube's engine been correctly adjusted, the governor would have taken care of the speed during the variation of pressure, and no runaway would have occurred.

H. W. BENTON.

Cleveland, O.

The letter appearing in the July 19 issue entitled "Runaway Engines," is, no doubt, a step in the right direction. The recommendations, however, are hardly adequate and by the expenditure of a few more dollars a complete installation can be made whereby the safety of both employee and equipment is made independent of the presence of mind of the engineer.

In the first place, not every man will rush forward and shut down an engine when it is running at a speed which is liable to burst the flywheel at any instant. Moreover, the chances are that, by the time the engineer reaches the stop valve the damage will have been done, especially if the increase in the speed of the engine has not been noticed immediately. It is obvious, therefore, that a mechanism for tripping the valve that will operate automatically will prove the best safeguard against accidents of this kind. Such a mechanism should not be dependent upon the regular engine governor for its operation; any device of this kind, if attached to the governor, would be of no greater value than the governor itself, as has been proved time and again.

An arrangement that has been found quite satisfactory is that of having an additional governor attached to the engine shaft by means of either a belt, chain or gears; although the latter method is not always advisable. The governor should be arranged so that it will trip a quick-closing valve when the engine reaches the upper speed limit. A common method of operating this system is by means of an electric contact connecting with a solenoid, which closes the valve. The necessary current can be obtained from batteries or the generator, if the engine be connected to one. A tripping lever should also be provided to shut down the engine in case the belt or chain breaks. Practical demonstrations of installations of this kind have shown that it is possible to have the steam entirely shut off within four seconds after the engine reaches the speed limit.

Such a system is not very elaborate, and there are few engineers who will not agree that it is a good investment.

EVERARD BROWN.

Pittsburg, Penn.

Weakness as Disclosed by Boiler Explosions

In several descriptions of boiler explosions that have appeared in recent issues of POWER, a point has been made of the fact that the metal had torn in the solid plate rather than at the joint or having

sheared the rivets. This fact was stated to indicate that the plate had been damaged or deteriorated to such an extent that it was weaker at the torn section than at the joints. It is a well known fact to those who have made a study of boiler explosions, that aside from the point of initial rupture, the breaks in a boiler structure, due to an explosion, do not follow the lines of least resistance, by any means. Frequently a line of rupture will take a girthwise course, although a cylindrical vessel is twice as capable of resisting rupture in this direction than it is longitudinally; this is without considering the strengthening effect of the tubes, which is an important factor always present in the horizontal tubular boiler.

The erratic nature of ruptures produced by boiler explosions is conclusive proof of the theory advanced by Zerah Colburn regarding the manner in which such explosions are produced.

This theory is in effect, that the initial rupture, (which, of course, occurs in a section that is least capable of resistance) produces a water-hammer effect in the boiler due to the sudden release of pressure in the steam space, and subsequent swelling of the contained water due to the instantaneous formation of steam bubbles throughout its mass. As is well known, ruptures due to water-hammer do not follow the lines of least resistance, but follow the direction of application of the force. Thus in a pipe line, a water-hammer usually blows out a cap at the end of the pipe or destroys a fitting; most of the failures being in a section that would have been the last to have failed under a steady application of internal pressure, and the same fact is true of boiler explosions. Therefore the nature or direction of ruptures produced by boiler explosions cannot serve to determine the probable relative strength of the different parts.

S. F. JETER.

New Haven, Conn.

Central Station versus Isolated Plants

It is evident from Mr. Marks' letter in the July 12 issue that he does not understand the theory of rate making for electric service. He considers it wrong that the small consumers should have to pay three or four cents per kilowatt-hour more than the average price, while the large consumers, who can afford to operate their own plants and consume but a small proportion of the total output, enjoy the special privilege of paying three or four cents less than the average rate. Mr. Marks errs in assuming that the price per kilowatt-hour should be made the same to all. As this misapprehension is shared by many and hinders a proper settlement of the rate question, the following may be timely:

It is recognized by the electrical interests that it is radically wrong to fix a uniform rate per kilowatt-hour. That which the central stations sell and which the consumers buy is service and not merely a certain number of units of electricity. This service consists of several factors, which vary greatly in their relative proportions; often the cost of energy consumed is small in comparison with some of the other factors. To illustrate this, suppose that a plant is to be installed to supply one hundred consumers with light. To determine the capacity of the plant, the average maximum demand of each consumer must be known. Suppose this to be five kilowatts per hour; then a plant of 500-kilowatt capacity must be installed. Some consumers will not require more than one kilowatt, while others may require 20 kilowatts. It is evident that the investment necessary to care for one may be 20 times that required for another. As soon as the plant is installed, the investment becomes subject to interest, taxes, insurance, depreciation, etc., which are practically constant regardless of the energy used by the consumers, and it is plain that each consumer should pay such a proportion of these charges as the investment made to take care of his needs bears to the total investment; he should pay this regardless of the kilowatt-hours used. This may be charged to "readiness to serve."

After the plant is in operation, there is a certain amount of work that must be done for each consumer, such as reading and caring for meters, keeping the records of his equipment, etc., and making out and collecting bills. All of these are practically independent of the energy consumed and may be termed a bookkeeping charge.

Finally, there is the cost of generating the power. This varies greatly, depending on the efficiency of the equipment and lines, the cost of fuel, labor, etc., but will be the same per kilowatt-hour for each consumer. If a consumer who has many lamps installed uses them but a short time each day, the amount of energy he uses may be no more than that of a consumer who has only a few lamps, but who uses them a much longer time. Suppose the "readiness to serve" charge amounts to $2.50 per month per kilowatt of installation, the bookkeeping charge is $1 per month and the cost of generating power is two cents per kilowatt-hour. If one consumer has lamps amounting to 10 kilowatts and consumes 500 kilowatt-hours per month, he should pay $10 \times \$2.50 = \$25$ per month for the "readiness to serve." He should pay $1 for bookkeeping charge and two cents per kilowatt-hour; a total of $36 per month, which is an average of 7.2 cents per kilowatt-hour. Another consumer having lamps amounting to only two kilowatts may use the same amount of energy, in which case he should pay $5 per month for "readiness to serve" charge; $1 bookkeeping charge; and $10 for energy; a total of $16, which is an average of 3.2 cents per kilowatt-hour. The average price paid by both is 5.2 cents per kilowatt-hour. Will Mr. Marks insist that it is just to charge both of the consumers at the same rate?

The public in general refuses to give the matter the study necessary to enable them to grasp the principles underlying the making of rates for electric service; hence much of the clamor about unjust discrimination.

I had charge of two plants where an effort was made to introduce this system of charging, but found it very difficult. One month each consumer, who would be benefited by the change, was billed on the "readiness to serve" schedule instead of the regular rate, and one business man came into the office and made a big stir about his bill. I tried to explain the schedule to him, but he would not listen and left, saying, "Young man, this is the last time you will ever get a dollar for keeping books for me." Thereafter he paid at the regular schedule, several dollars more per month than the "readiness to serve" plan would have cost him.

When a central-station management approaches an isolated plant, the cost of serving the consumer having the isolated plant is usually carefully figured and a rate offered that will place the central station on the safe side.

Does Mr. Marks suppose that the central station hustles to get a large consumer, knowing that it will have to hustle for many small consumers in order to make up for what is lost on the large one?

A business concern hustles for the business that pays, and usually avoids the unprofitable customer. It costs more per kilowatt-hour to serve small consumers than large ones; hence they must be charged more. I admit that there is room for a readjustment of rates, but do not believe it should be along the line of a uniform rate to all.

G. E. MILES.

Salida, Cal.

Boiler Flanging

A prominent manufacturer of fire-tube boilers inserts the following clause in his specifications: "Hydraulically flanged heads are not to be used under any circumstances. Heads are to be machine-flanged by the spinning process." The writer does not see why hydraulically flanged or pressed heads are not superior in every way to spun heads. In the first place, the pressed heads of each size are of uniform diameter, making it easy to properly fit the courses to the heads. This uniformity is of consider-

able importance where a number of boilers of one size are built together, because, owing to the difficulty of keeping the shell plates and heads separate for each individual boiler in the shop, the person laying out the work in the case of the spun heads of varying diameter, usually endeavors to strike an average for the lot that will best suit all; this usually results in a poor fit for some.

Again, the flange on a pressed head is thicker than the plate from which the head is made; this is a desirable feature and is due to the upsetting of the metal in the flanging process. With the spun head the flange is usually slightly thinner than the plate from which the head is made. The surface on the flange of a pressed head is smoother than that on a spun head, the latter often containing grooves made by the rolls of the spinning machine; this smoothness is important in making tight-head seams without undue caulking. Furthermore, the surface of a pressed flanged head is usually straighter than one which is spun, permitting the use of uniform tube lengths without the necessity of springing the heads into position. The value of this feature after the boiler is constructed is only thoroughly appreciated by the workman who has to put in a new set of tubes, the owner being affected only by the increased cost of the work.

With the multiplicity of diameters called for at the plate mill, the spinning process is by far the most practical for general purposes, but it would seem that a boiler manufacturer who intends to have complete equipment for the manufacture of high-grade boiler work would flange his boiler heads on a press. It seems that the condemnation of the hydraulically flanged head is a subterfuge, intended to mislead the purchaser who is not well informed upon the subject.

F. O. JONES.

Boston, Mass.

Patching Boilers

The proper method of applying patches to boilers is occupying the attention of POWER readers at the present time, the subject being viewed principally from the theorist's standpoint, but with scant consideration for the boilermaker who is to actually do the work. It is proper that safety be given first consideration, and it is aiding this cause to see that the prescribed methods of repair are thoroughly practical.

The majority of patches that are placed on boiler shells are of such moderate dimensions that their shape and the exact spacing of the rivets are not a material factor affecting the strength; in such cases the selection of a method that is most likely to give permanent repair is of paramount importance. I feel perfectly safe in saying that the trouble experienced with patches in 99 cases out of 100 is due to inability to keep them tight; and this lack of tightness is not caused by inability of the patches to resist rupture due to strains caused by the steam pressure. From the standpoint of strength alone, a patch should always be placed on the inside of the shell plate; but there are other important features to be considered. For example, assume that a patch is to be placed on the outer course of a boiler, as illustrated in Fig. 1; it is evident that to facilitate its application such a patch should be placed inside the course, because the ends of the patch can be readily scarfed before

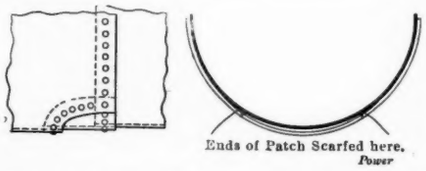


FIG. 1. PATCH APPLIED ON OUTER COURSE

it is placed in position. It will then fit smoothly between the shell plates, where they lap over the patch at the girth seam, and a tight joint is assured. If such a patch were placed on the outside of the shell it would be necessary to scarf the edges of the course at the girth seam; this would be difficult and the girth edge of the patch would have to be raised out of level at the corners, which would increase the difficulty of making a tight and lasting job. If the patch were located on an inner course of the boiler, as illustrated in Fig. 2, then it would be necessary to place it on the outside of

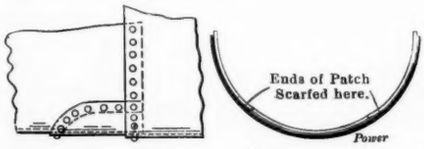


FIG. 2. PATCH APPLIED ON INNER COURSE

this course for the same reasons as just given, except that they would apply with greater force, because the scarfing of the corners of the inner sheet, which would become necessary with a patch applied internally, would be a very awkward operation. With a patch of considerable size, where ample room is afforded for working inside the boiler and the extent of the patch lessens the difficulty of having one edge out of level, these points do not apply with such force.

Frequently a patch is required on a return-tubular boiler at the point where the blowoff enters. To facilitate draining the boiler it would be desirable to locate such a patch on the outside of the course, but when the patch must include the head seam, as is generally the case, it should be placed inside the shell to permit the making of a tight fit. Usually the deciding factor as to placing a patch on the inside or the outside of a sheet, where it is to comprise a portion of a girth seam, depends upon whether the boiler sheet, to which it is attached, is an inner or an outer course. Where the girth seam is not involved, a patch can usually be put on with equal ease on either side of the sheet.

I do not believe in the generally accepted opinion that the slight depression caused by placing a patch on the outside of a sheet materially increases the liability of scale to collect at such a point. It would seem to be equally consistent to condemn a cylindrical form of heating surface for the same reason and demand a perfectly flat surface, or one convexed on the water side. From the stress that some engineers and inspectors lay on this point it would seem that a whirlpool action, drawing scale from all portions of the boiler, was expected to occur over every slight depression in a boiler plate. The liability of scale to collect on a patch is usually due to causes that produced the initial collection which made the patch necessary.

J. E. TERMAN.

New Haven, Conn.

Unusual Boiler Behavior

In the August 2 issue, Mr. Haeusser describes a boiler experience which he terms unusual. The behavior of the boilers may have been unusual but no boiler could have acted otherwise under the circumstances.

The proper thing for the watchman to have done would have been to fire No. 1 boiler first, because it was the colder of the two boilers. Nevertheless, there was nothing wrong with firing No. 2 first, if he had only shut off the draft on that boiler until the gage showed No. 1 to be steaming well. As it was, No. 1 boiler was being fired at a rate that might be called forcing before the water had reached the boiling point.

When the engines were started the steam pressure dropped because No. 2 boiler could not carry the load alone. Mr. Haeusser pulled the safety-valve chain on the inactive boiler. This lowered the pressure and started action within the boiler.

I have seen boilers behave in the same way. In a battery of boilers some were used all night while the others had the fires cleaned and dampers closed, but the fires were not kept hot enough to maintain the temperature of the water at the boiling point corresponding to the pressure. When the load was thrown on in the morning there was a scramble for steam until all the boilers were hot enough. There was no film of cold air present which might be blamed for the trouble and no safety-valve chains to pull, so we just had to get busy and get the boilers hot.

R. MCLAREN.

Berlin, Ont.

The Plotting of Curves*

By F. R. Low

When it is necessary to determine the efficiency of any type of boiler joint it is a great convenience to be able to refer to a diagram which will tell the whole story. Such a diagram is presented in this article and by referring to the familiar expansion curve of the indicator diagram it is shown how these diagrams are plotted.

Mr. Jeter in his lecture on "Riveted Joints," delivered to the club some time ago, described a diagram devised by himself which was in its first form like Fig. 1.

If, for example, the one-inch rivets, for which this particular diagram is drawn, are spaced or "pitched" 2.5 inches apart in an 11/32-inch plate, the horizontal line representing that pitch intersects the vertical line representing that plate thickness at *B*. This point of intersection lying in the lower of the three fields into which the plane is divided by the diagram indicates that the joint may be expected to fail by the tearing apart of the plate as indicated by the sketch in that portion of the field.

If a wider pitch is used there will be fewer rivets to stand the pull, greater stress on each rivet, and more probability that the rivet will crush the plate in front of it, as shown by the sketch in the upper left-hand field. The intersection of the lines representing 3-inch pitch and 11/32-inch plate falls in this field at *A*, indicating that a joint made with 1-inch rivets and these proportions would be expected to fail by crushing. If the plate thickness be increased to 3/8 inch, however, it will not crumple itself nor crush the rivet but will shear it off, as shown by the sketch in the upper right-hand field, in which field the intersection of the lines representing 3-inch pitch and 3/8-inch plate falls at *C*.

If the efficiency of a given joint is wanted it is necessary only to see from the diagram in what manner it may be expected to fail, and compare the force required to produce failure in that way with that required to part the solid plate. With complicated joints which may fail in six or more different ways this diagram, modified to include all possible methods of failure by the simple expedient of limiting the lengths of the lines, as Mr. Jeter explained, becomes of great

*A note to the Modern Science Club, discussing Mr. Jeter's paper on riveted joints.

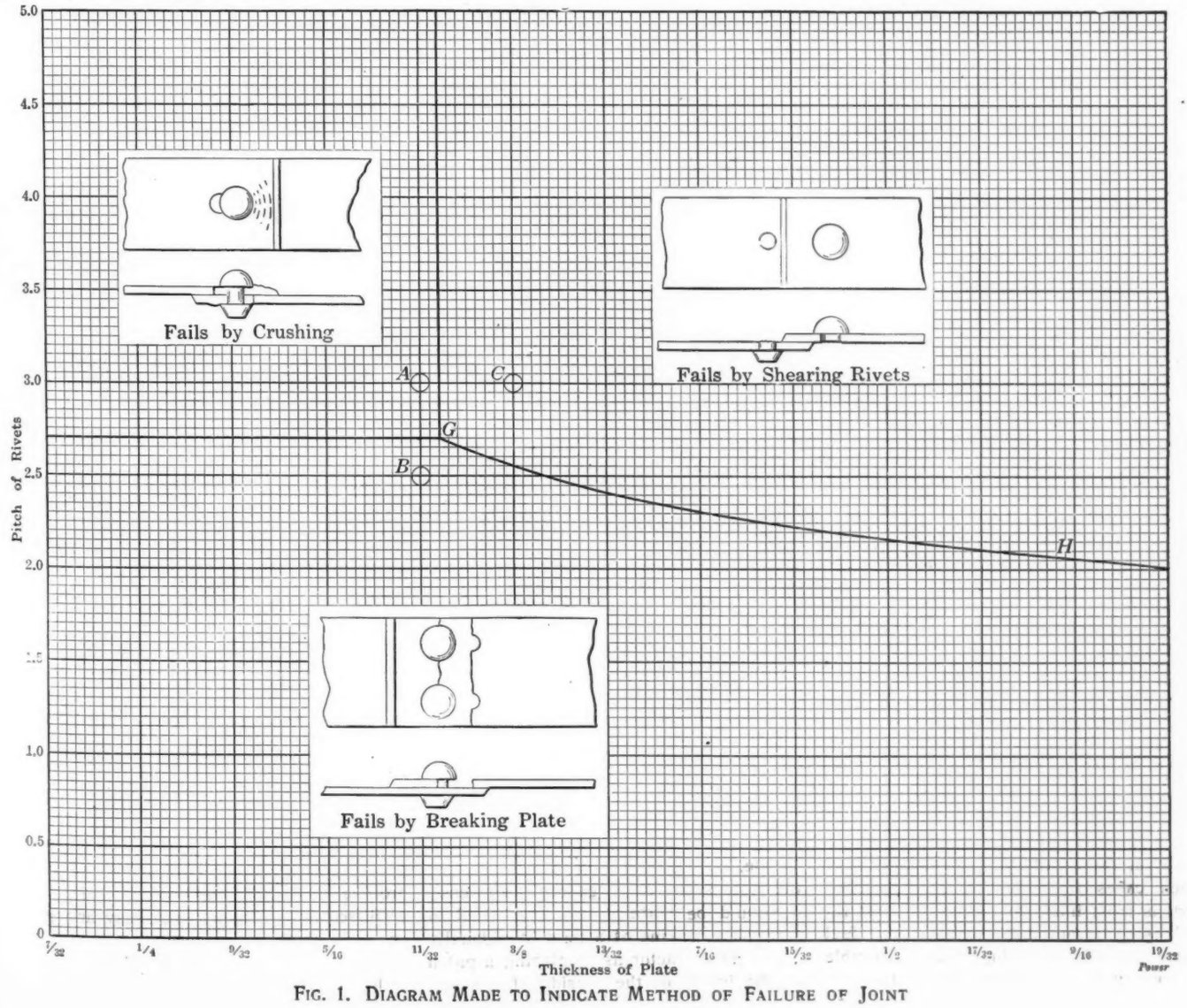


FIG. 1. DIAGRAM MADE TO INDICATE METHOD OF FAILURE OF JOINT

assistance and enables one not only to arrive quickly at results which would otherwise require extensive calculation, but to see at a glance the limitations and tendencies of the case.

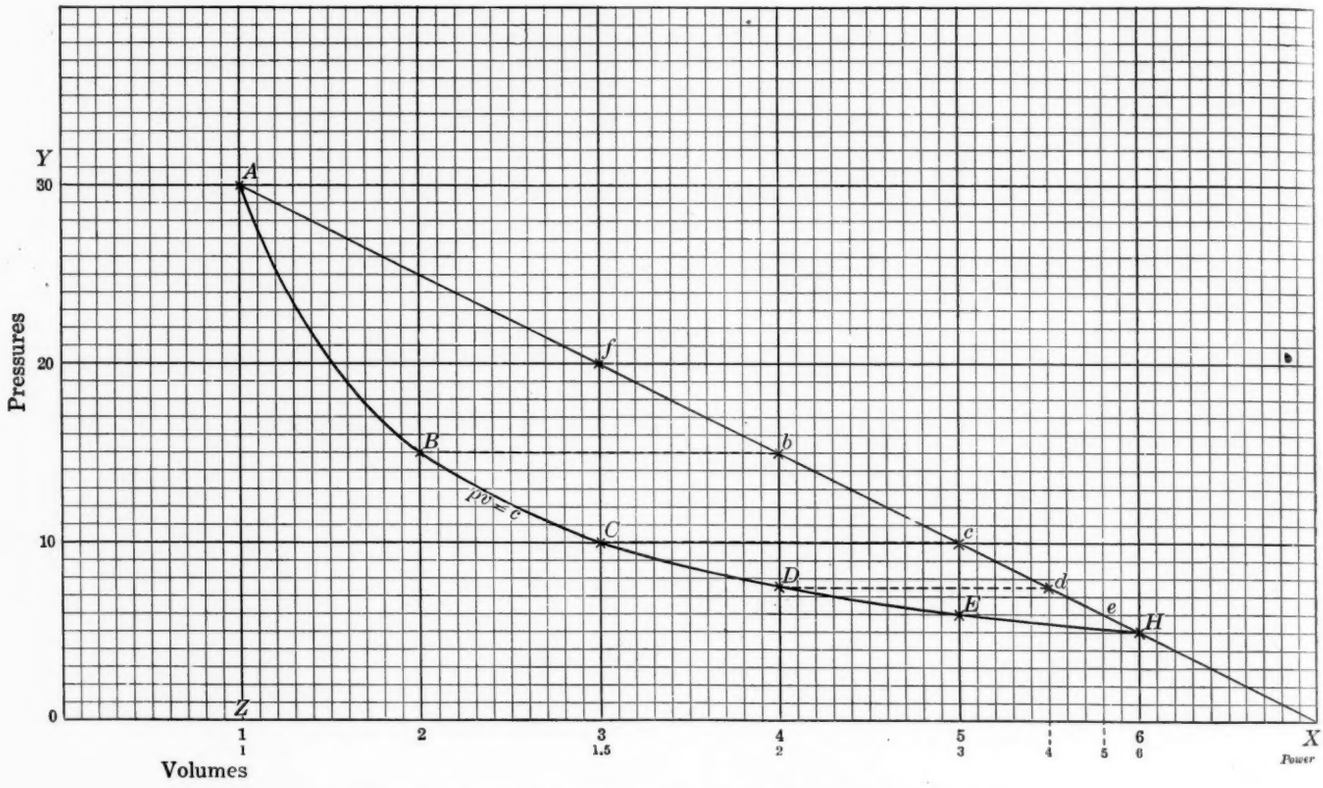


FIG. 2. ORDINARY INDICATOR-DIAGRAM CURVE AND THE STRAIGHT-LINE RECIPROCAL

One of these diagrams has to be plotted for each size of rivet in each type of joint, and for each different assumption that is made of tensile shearing and crushing strengths. This would be a good deal of a task if the line *G H* separating the plate-failure from the rivet-failure field had to be plotted point by point and traced in the curvilinear form which it has in Fig. 1. Mr. Jeter remarked, however, that by using *the reciprocals* of the horizontal distances in plotting them instead of the distances themselves, the curve became a straight line, and, having the commencement of it at *G*, it was necessary only to locate one other point upon it and put in the whole line with a straight-edge. Diagrams made in this way for all sizes of rivets and many types of joints with directions for making them were published in POWER for January 5, 1909.

It is this particular feature of Mr. Jeter's lecture that I thought it might be interesting to expand; and the object of this note is to show by referring the process to the familiar expansion curve of the indicator diagram more particularly how this is done.

The curve in Fig. 2 is that ordinarily applied to the indicator diagram, where the product of the distances of any point upon the curve from the axes is constant. For instance, the point *A* is 10 spaces from the *O Y* axis and 30 spaces from the *O X* axis, and

$$10 \times 30 = 300.$$

The point *B* is 20 spaces from the *O Y* axis and 15 spaces from the *O X* axis, and

$$20 \times 15 = 300.$$

In a similar way if the distance horizontally of any point whatever upon the curve be multiplied by the vertical distance from the line *O X* the product will be 300. It is this characteristic of the curve which makes it representative of that mode of expansion in which the product of the pressure, represented by vertical distances, and the volume, represented by horizontal distances, is constant.

To lay out the curve set off the *equally spaced* ordinates 1, 2, 3, 4, 5, 6 and upon ordinate 2, where the volume is supposed to be doubled, lay off one-half of the initial pressure; on ordinate 3 lay off one-third; on ordinate 4, one-fourth, etc.

For most purposes of the steam diagram it is better to use the curve and to keep the horizontal distances proportional to the volume or piston movement; but if it were desired simply to make a diagram from which could be measured the volume for any pressure or *vice versa* the curve could, by the simple expedient of using *the reciprocals* of the volumes instead of the volumes themselves, be reduced to a straight line. Such a straight line would be *A H*, Fig. 2, and the point indicating the pressure for two volumes

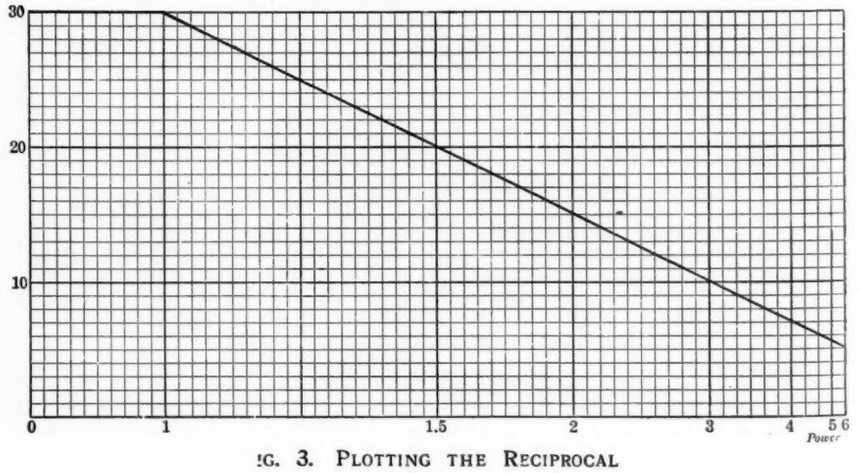


FIG. 3. PLOTTING THE RECIPROCAL

would lie upon it at *b*, at the same hight as *B*, because, of course, the same hight must be used to indicate the same pressure upon the straight as upon the curved line. The ordinate or vertical line, upon which this point falls now becomes the ordinate for two volumes instead of four, as it was with the curved line, and has been marked 2 in the lower line of numberings for the ordinates. The pressure for three volumes would lie upon the straight line at *c* at the same hight as *C* upon the curve, and the ordinate upon which it lies is marked 3 in the lower line of values. The ordinates for four and five volumes are similarly located. As the straight line *A X* passes through the original point for six volumes its ordinate remains the same.

The point *b* indicating the pressure for two volumes must be one-half as high above the base line *O X* as is the point *A* indicating the pressure for one volume; therefore, it must be one-half the distance of *A* from the intersection *X* of the line *A X* with the base. From *X* to *Z* is 60 spaces; the ordinate for two volumes is one-half of this or 30 spaces from *X*. The ordinate for three volumes is one-third of this or 20 spaces from *X;* the ordinate for four volumes is one-fourth of 60 or 15 spaces from *X;* the ordinate for five volumes is one-fifth of 60 or 12, and the ordinate for six volumes is one-sixth of 60, or 10 spaces, from *X*. In each case the spacing is proportional to the *reciprocal* of the volume, i.e., 1 divided by the volume. The ordinate for 1.5 volumes for example, would be located

$$\frac{1}{1.5} \times 60 = 40$$

spaces from *X*, and the pressure would be that represented by a vertical distance of 20 spaces, as is indicated by the intersection of the line *A X* with that ordinate at *f*. The ordinate for 1 volume would be that under *A*, as before, for

$$\frac{1}{1} \times 60 = 60$$

The diagram will now be like Fig. 3. The spacings of the ordinates instead of being uniform as in Fig. 1, or as the spacing which produced the curve in Fig. 2, are irregular, growing closer as they go to the right. They are not, however, logarithmic like the spacing upon a slide rule. In logarithmic spacing the distance from 1 to 2 will be the same as from 2 to 4, 3 to 6, 4 to 8, etc., and the distance from 1 to 3 the same as from 2 to 6, 3 to 9, 4 to 12, etc. In Fig. 3 the distance from 1 to 2 is 30 spaces; from 2 to 4, 15 spaces, etc.

When the ratio of expansion is constant a steam engine uses a fairly constant amount of steam for each indicated horsepower-hour plus a constant quantity required to overcome cylinder losses.

Suppose the internal losses of a certain engine to be 30 pounds per hour, and suppose it to use 20 pounds additional for each horsepower developed. The steam used per hour will be as follows:

Lb. per H.P.	20	20	20	20	20	20
H.P.	0	1	2	3	4	5
	00	20	40	60	80	100
Cyl. loss	30	30	30	30	30	30
Total steam	30	50	70	90	110	130

In Fig. 4 lay off on the vertical axis *O Y* a scale of total steam consumption in pounds per hour, and on the horizontal axis *O X* a scale of horsepowers. Then set off on the ordinate for 0 horsepower 30 pounds at *a;* on the 1 horsepower ordinate 50 pounds at *b*; on the 2-horsepower ordinate 70 pounds at *c*, etc., and it will be apparent that the points lie in a straight line, for the increase per horsepower is regular, so that *c* is just as much above *b* as *b* is above *a* and for equal horizontal distances the rises (as *d b* and *e c*) will be equal. The total rise above the level of the point *a* will be 20

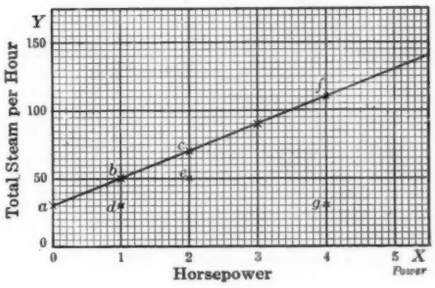


FIG. 4. STEAM CONSUMPTION PER HOUR

times the horsepower. For instance, the line *f g* giving the increase in consumption by the imposition of 4 horsepower is

$$20 \times 4 = 80$$

pounds, and the total consumption per hour is

$$80 + 30 = 110$$

pounds.

Calling the total steam per hour *S*,

$$S = 20 \times HP + 30$$

The total steam per hour is the hight on the *Y* axis; the horsepower is the length on the *X* axis. The numerical quantities can have any value at all and the line will remain a straight line so long as they remain constant.

Any relation, therefore, between two varying values *x* and *y* which can be expressed by the formula

$$y = a x + b$$

will be represented by a straight line, and the instant the mathematician sees a formula of this sort he knows that if he plots *x* and *y* against each other the result will be a straight line.

The final term (*b* in this case) may be plus or minus or absent altogether without changing the line except as to the origin. If it is absent the line will commence in the lower left-hand corner at the intersection of the *X* and *Y* axes. Let us see what kind of a formula we have to deal with in the case of the riveted joint.

To find the force required to tear the plate apart as in the lower sketch in Fig. 1, the tensile strength per square inch must be multiplied by the area left between the rivet holes, the area *a b c d*, Fig. 5. This area is the product of the thickness *t* and the length *a b* which is the difference between the pitch and the diameter of the driven rivet. Calling the tensile strength in pounds per square inch *T*, the force necessary to break an element of the joint by parting the plate would be

$$t\,T\,(P - d).$$

The force necessary to shear a rivet is the cross-sectional area of the rivet in square inches multiplied by the shearing strength per square inch, or

$$0.7854\ d^2\ s.$$

Making these equal to each other,

$$t\,T\,(P - d) = 0.7854\ d^2\ s.$$

Substituting the values of 52,000 for *T* and 42,000 for *s* and solving for *P* produces the following:

$$52{,}000\ t\ (P - d) = 0.7854 \times 42{,}000\ d^2$$

$$P - d = \frac{0.7854 \times 42{,}000\ d^2}{52{,}000\ t}$$

$$P = \frac{0.7854 \times 42{,}000\ d^2}{52{,}000\ t} + d$$

$$P = \frac{0.6\ d^2}{t} + d$$

If a table or diagram of the values of *P* for a given diameter *d* of rivet were being made, the expression 0.6 d^2, which is to be divided by *t*, would remain the same all through the series of calculations for that size of rivet. Represent that constant by *a*. The *d* which is added to the quotient but not divided by *t* also remains constant.

Substituting *a* for 0.6 d^2 gives

$$P = \frac{a}{t} + d$$

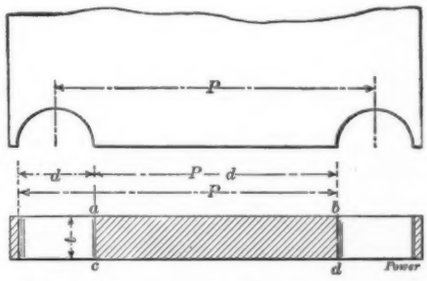


FIG. 5. SECTION OF PLATE BETWEEN RIVETS

which can as well be written

$$P = \frac{1}{t}\,a + d$$

This, it will be seen, is a straight line formula, *P* the hight on the *Y* axis equals $\frac{1}{t}$ of the distance on the *X* axis, multiplied by a constant with a constant added to it. It is evident beforehand, therefore, that if $\frac{1}{t}$, *the reciprocal* of the thickness, is plotted against the pitch, a straight line will result.

Let us start a plot for a ⅛-inch rivet. If $d = ⅞ = 0.875$.

$a = 0.6\ d^2 = \frac{6}{10} \times \frac{7}{8} \times \frac{7}{8} = \frac{294}{640} = 0.446$

Then the operation will be

$$P = 0.446 \frac{1}{t} + 0.875$$

Thickness.	*Reciprocal of the Thickness.
¼	¼ = 4
5/16	5/16 = 3.2
3/8	3/8 = 2.67
7/16	7/16 = 2.285
½	½ = 2
9/16	9/16 = 1.777
5/8	5/8 = 1.6
11/16	11/16 = 1.45
¾	¾ = 1.33
13/16	13/16 = 1.23
⅞	⅞ = 1.14

*To get the reciprocal of a whole number write the number as a denominator with 1 as a numerator. To get the reciprocal of a common fraction turn it upside down.

The purposes of this diagram are best served by having the *Y* axis at the right instead of at the left-hand end of the *X* axis. Lay off the scale of pitches on the vertical axis *O Y* in Fig. 6, and the scale of plate thicknesses along the line *O X*. First divide the line into a regular scale, as by the black-faced figures. I have given two of the main divisions to each unit, making each of the finer subdivisions equal to 1/20, or 0.05. Setting off the values of $\frac{1}{t}$ for the plate thicknesses we wish to consider, on this scale we find that the reciprocal of ¼ which is $\frac{1}{\frac{1}{4}} = 4$ falls upon one of the main divisions at *a;* the reciprocal of 5/16 is $\frac{16}{5} = 3.20$, and this is four of the smaller divisions beyond 3 at *b*, because such division represents 0.05 and it takes four of them to represent 0.20. The reciprocal of ⅜ is $\frac{8}{3} = 2.67$, and the ordinate for this is at *c*, three of the 0.05 divisions and 0.02 more estimated to the left of the 2.5 mark. In the same way the ordinates for the reciprocals of the rest of the thicknesses are located.

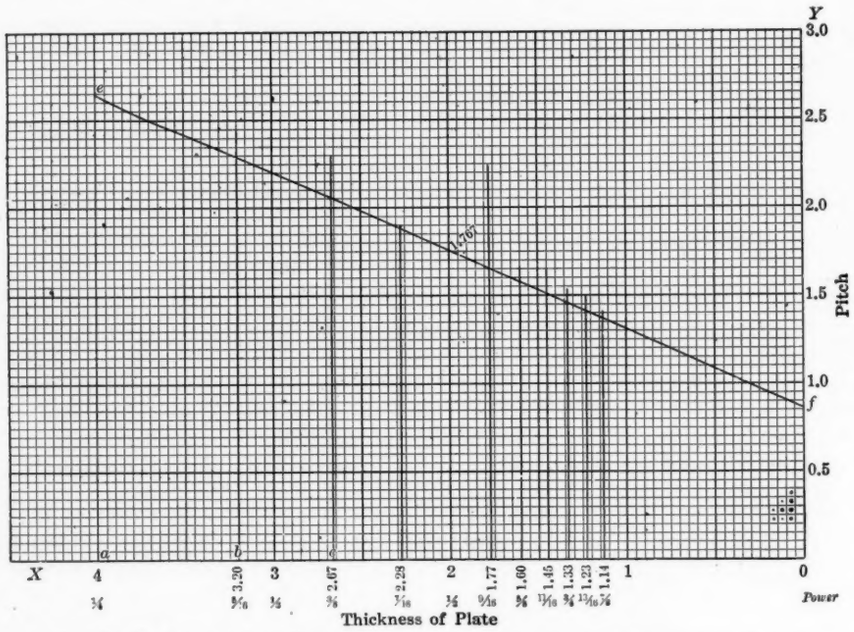


FIG. 6. RECIPROCAL OF THICKNESS OF PLATE PLOTTED AGAINST THE PITCH

The formula is

$$P = 0.446 \frac{1}{t} + 0.875$$

For a ¼-inch plate $\frac{1}{t} = 4$.

$$0.446 \times 4 + 0.875 = 2.659.$$

For a plate of zero thickness $\frac{1}{t} = 0$, and $0 + 0.875 = 0.875$.

Setting these values off upon the 4 and zero ordinates respectively, the points *e* and *f* are located and the line *e f* may be drawn with every confidence that the values for all of the other ordinates will fall upon it.

For instance, let us calculate that for ½-inch plate. The pitch for this thickness is

$$P = 0.446 \times \frac{1}{t} + 0.875 = 1.767.$$

If $t = ½$, $\frac{1}{t} = 2$, and

$$P = 0.446 \times 2 + 0.875 = 1.767.$$

The reason that the use of the reciprocal of the volumes reduced the pressure-volume curve to a straight line is easily seen. The formula $p v = K$ (here $K =$ a constant) transposes to

$$p = \frac{K}{v}$$

which produces an hyperbola, but if the reciprocal of v be used as a multiplier instead of v itself as a divisor, the formula becomes that of the straight line

$$p = \frac{1}{v} K$$

If you want to multiply two numbers together you can add their logarithms. The sum will be the logarithm of the product of the two numbers.

For example, multiply 2749 by 3.1416.

No.		Logarithm.
2749	=	3.4391747
3.1416	=	0.4971509
8636.3	=	3.9363256

You find in a table of logarithms the logs of the two given numbers. Add them, find the new logarithm made by their sum in the table and the corresponding number is the product.

It follows that if you wanted to multiply a number by itself a given number of times you would simply have to add its logarithm together that number of times. For example, to raise 128 to its cube or third power

128 =	2.1072100
128 =	2.1072100
128 =	2.1072100
2097152 =	6.3216300

Easier than this would be to multiply the logarithm by 3, which would, of course, be the same as adding three of them together.

To raise a number to a given power multiply the logarithm by the exponent of that power, the product will be the logarithm of the given power.

The formula, $p v = Constant$, gives the curve *a b*, Fig. 7. The formula, $p v^2 = Constant$, gives the curve *a c*. The formula, $p v^{0.5} = Constant$, gives the curve *a d*.

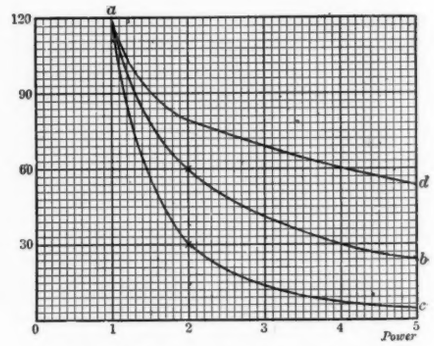


FIG. 7. PRESSURE-VOLUME CURVES

The pressures for the various volumes are gotten as follows:

For the case of $p v = Constant$:

v	v^2		p		Constant.
1	1	×	120	=	120
2	4	×	30	=	120
3	9	×	13⅓	=	120
4	16	×	7½	=	120
5	25	×	4.8	=	120

The volumes corresponding to the ordinates of Fig. 7 are given in the first column and their squares in the second. The product of these squares and the pressures at the corresponding volume will be 120. The pressures in the third column are therefore found by dividing 120 by the squares of the volumes.

For the case of $p v^{0.5} = Const$

The 0.5 or ½ power of a number is its square root. In the second column below are the square roots of the volumes, which can be found in a table of squares and roots to be found in most engineers' handbooks. The pressures are

found by dividing the constant 120 by these roots.

v	$v^{0.5}$		p		Constant.
1	1	×	120	=	120
2	1.414	×	84.9	=	120
3	1.732	×	69.3	=	120
4	2	×	60	=	120
5	2.236	×	53.7	=	120

By varying the exponent you can make the curve go where you please or fit any expansion line as long as it is this kind of a curve. The exponent in $pv =$ *Constant,* is, of course, unity. When the exponent becomes some such number as 1.69 it would be difficult to raise the volume to that power without the logarithms.

The formula then is

$$pv^n = K \text{ and } p = \frac{K}{v^n}$$

Since multiplication is performed by adding the logarithms, the inverse operation, division, is performed by subtracting them; then

$$\log p = \log K - n \log v,$$

which is the straight-line formula again with a simple transposition of its terms.

If, therefore, the volumes and pressures are scaled logarithmically, the curve will become a straight line and may be laid off with a ruler between two determined points and intermediate points measured on the logarithmically-spaced ordinates. Paper ruled logarithmically for this purpose is procurable from dealers in draftsmens' supplies.

The Irish Fireman and the Engineer

By E. A. Dixey

It's hard to get the best of an Irishman when it comes to a duel of wits, and this story, told me by my friend "Mac," third assistant engineer of a transatlantic liner, corroborates the foregoing statement.

"In our boats all the men are known by number. It's not like it used to be in the old tramps I sailed in: 'Here, Murphy, fire up;' or 'Swenson get some more coal on the after plate;' but it's '56 do this,' and '94 do that.' It's only one here and there whose name we know.

"I was standing with my back against the bulkhead one night on the twelve to four watch. Everything was going lovely, a fireman's fair wind, that is to say, a light breeze dead ahead, plenty of steam, good coal and nothing under the stars to bother either me or the men under me.

"Our crew was composed of Irish, Liverpool Irish at that—you know them as well as I do—and one of the trimmers, Murphy by name and 87 by number, was taking a blow under the ventilator at my elbow. Said I:

"'87, your name's Murphy, isn't it?'

"'Yes, sorr.'

"'You come from Liverpool, don't you?'

"'Oi do that, sorr.'

"'Well, I heard a story about a man of the name of Murphy and, by the way, he came from Liverpool, too. Perhaps he's a relation of yours. He died and off he went straight as a dart for heaven, and at the door he was stopped by an archangel with a flaming sword, while Saint Peter pushed back a little sliding wicket, stuck his head out and said:

"'What's your name?'

"'Murphy, sir.'

"'What are you to trade?'

"'Marine trimmer, sir.'

"'Where do you come from?'

"'Liverpool, sir."

"'[illegible]t the gate, Gabriel, we have no use for *him* here,' said Saint Peter.

"Well, with that, Murphy, feeling kind of chilled, owing to the exposed position of the gate of heaven, decided that he would have a try at getting into another and warmer place, which he had always been told was much easier of access. So down he dropped and in a few moments found himself at the gate of hell with 'Old Nick' himself peeking at him through the bars. Murphy took heart as he saw the friendly way the devil was looking at him, and said to himself: 'Shure, he's one of my own trade, but I'm no more than an apprentice alongside of him at the coal-burning business.'

"Well, the devil says:

"'What do you do for a living, young man?'

"'My name's Murphy and I am a marine trimmer, sir.'

"'Where do you come from?'

"'I come from Liverpool, sir,' says Murphy.

"'Is *that* so?' says the devil, in a most sarcastic tone; 'then we've got no use for *you* here.'

"'But, sir,' said Murphy, 'you're not going to send me back to *Liverpool,* are you, sir?'"

"It's a good yarn, Mr. 'Mac,' and as soon as I get a bit more coal out of the bunker, sorr, oi'll tell ye wan about another Liverpool maan, but *his* name was Hennessey,'" said 87.

So I waited till he came out of the bunker again and he started:

"'The man oi'm tellin' ye about was a Liverpool fireman av th' name av Hennessey, and he died just the same as the man ye wor tellin' about, and aff he goes for heaven, hangin' onto the tail av a shootin' star, and by and by he drops aff just at the gate of heaven, stubbin' his toe on the gate post. The noise brings Peter to the gate in a hurry, and he sings out kind av short and snappy:

"'Who are you, phawt's yure name and phwere do yez come from?' all in th' wan breath. And Hennessey, seeing th' jig was up, answers much in the same way:

"'Me name's Hennessey, marine fireman, from Liverpool,' and as he turns to go he hears Peter sing out:

"'Go to hel——p fire down below, they can prob'ly use the loikes o' you.' Th' p in help wa' a long way afther th' l; ut's the only kind o' profanity they're allowed up there.

"So down he goes, sits down on a pile o' clinkers just outside the dure o' hell, tears a bit aff his sweat rag to tie round his stubbed toe and just then he hears a voice behind him and as he turns he sees the divil lookin' friendly loike through the bars and he says to Hennessey:

"The top o' the mornin' to ye me friend, what's your name?"

"And Hennessey said: 'Me name's Hennessey, oi come from Liverpool, and oi'm a marine fireman to trade, is there a chance for a job?"

"'Come right in,' says the devil, as he swings the dure open; 'come right in and take them four fires in the wing.'

"Well, Hennessey steps inside and over to the wing, and bein' a good man he wants to get a look at his fires. There wor plenty of firing tools, slice bars, prickers, rakes and shovels but high or low he couldn't find a hand rag, and he spent a good deal of time looking for one.

"At last the devil comes over to him and says, sarcastic: "Oi hired ye for a *fireman,* not for a sojer. Now get busy wid yer firin'!'

"'All right, sir,' says Hennessey, 'but I can't fire without a hand rag and I can't find one, so if you want me to get ahead, dig up the rag.'

"With that the divil says: 'Ut's a hand rag ye want, is it. D'ye see that box over in th' corner? Open ut and ye'll find a lot of old chief ingineers in ut, they make damned good hand rags, and be gob it's all they're fit fer.'" And 87 went into the bunker again.

When connecting boilers, exercise great care. See that the pressures on the steam gages agree before attempting to open the steam valves. Open the valves very slowly, and, if provided with small bypass valves use these before opening the large valves. Never connect boilers unless the steam pressures agree. Many explosions have been caused by neglect of these precautions.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Heating Feed Water with Live Steam

If the exhaust steam is not sufficient for heating the feed water in a feed-water heater up to 210 degrees Fahrenheit, is it advisable to add live steam for the purpose of reaching the above temperature, or is it more economical to permit the water to enter the boiler at a varying temperature which is sometimes as low as 160 degrees?

J. M. D.

There is no fuel economy in heating feed water with steam drawn from the boiler which the water enters. But there are several good reasons why it is good practice to heat the feed water to the temperature of the steam in the boiler before it is pumped in, chief among which is that by so doing most of the impurities in the water are deposited in the heater instead of in the boiler.

Where the feed pipe passes through the front head to the rear, crosses and turns downward, discharging the water near the shell, the feed water will be so near the temperature of the water in the boiler that no injurious stresses will be set up by unequal expansion of the shell due to local temperature differences.

Pressure in Cylinders

With the throttle open, what pressure per square inch will there be in the cylinder with 100 pounds boiler pressure?

C. F. W.

In order to compensate for friction, heat losses, etc., 85 per cent. of the boiler pressure is commonly taken as the maximum cylinder pressure. This would give 85 pounds pressure per square inch. If the engine does not move and the throttle is open, pressure will increase in the cylinder until it equals that in the boiler, but with the engine moving, 85 per cent. of the boiler pressure may be taken as a fair average of the maximum cylinder pressure.

Raising Steam Pressure

Is it advisable to raise steam pressure rapidly in a locomotive type of boiler?

J. A.

No. As the water absorbs heat slowly, raising the steam pressure rapidly will have a tendency to injure the sheets by unequal expansion, as can be observed by placing the hand on the water leg of a boiler of the firebox type in which steam is being generated from cold water; it will be noticed that the leg of the boiler at the bottom will be cold, while at a distance of perhaps 12 to 18 inches above the sheet will be hot, thus producing an unequal expansion.

Heating Feed Water

How much good does it do to heat the feed water before it goes to the boiler?

J. C. J.

To be made into steam at working pressure the water must be heated from the temperature at which it comes to the boiler room to that of the steam which goes to the engine. All of the heat which goes to the water in the boiler must come from coal burned in the furnace and the more heat there is in the water when it goes into the boiler the less will have to be put into it to make steam. It has been found that there is a saving of 1 per cent. in fuel for each eleven degrees that the feed water is heated before it goes to the boiler.

Combustion

What is combustion, and is air necessary?

J. L. F.

Combustion is burning. It is a rapid combination of oxygen and whatever is burned and is usually spoken of in connection with the fuel used in steam making. As air furnishes the oxygen it is necessary for the process.

Pitch of Steam-pipe Line

Which way should a steam line have a fall, toward the boiler or toward the engine, and why?

F. A. C.

The line should pitch toward the engine.

Stresses in Boiler Seams

Why are the stresses in the longitudinal seams of a boiler greater than on the circumferential seams and why are the longitudinal seams double riveted?

F. A. L.

The force tending to tear the boiler apart along its length is for each inch of length double that tending to separate the shell endwise, and to resist the same pressure the longitudinal seams must be twice as strong as the circumferential ones.

Effect of Improper Air Supply

What is the effect upon combustion if too little or too much air is supplied through the fire?

J. W. A.

If the supply of air is so restricted that not enough is supplied, each atom of carbon will unite with but one atom of oxygen, forming carbon monoxide, which yields only 4500 heat units per pound of carbon consumed. In order to have perfect combustion sufficient air must be supplied so that two atoms of oxygen can unite with one atom of carbon, forming carbon dioxide. Under these conditions one pound of carbon will furnish 14,500 heat units.

Ordinarily, with the fire in proper condition too much air cannot be supplied through the fire. If, however, the fire is thin or has holes in it so that too much air is admitted, the air will cool the gases and carbon monoxide will be formed.

Waste of Fuel at the Safety Valve

What is the estimated waste of coal each minute the safety valve is open?

O. R. D.

In a 3-inch safety valve the estimated waste is 15 pounds of coal per minute.

Location of Fuses

What is the Underwriters' rule for the location of fuses in electric circuits?

R. B.

Fuses must be used at every point where the size of wire is reduced, unless the fuses protecting the larger wire are small enough to protect the smaller wire.

Safe Working Pressure

What is the safe working pressure of a 54-inch boiler, 5/16-inch plate with butt double-strap joint treble riveted; assumed tensile strength of sheet 60,000 pounds per square inch?

V. B. H.

The safe working pressure of a boiler shell, or drum, is expressed by the formula,

$$\frac{Ts \times t \times \%}{R \times F} = W.P.$$

in which

$W.P.$ = Working pressure,
Ts = Tensile strength of material,
t = Thickness of plate,
$\%$ = Efficiency of joint,
R = Radius of shell,
F = Factor of safety.

With the efficiency of the joint 87 per cent. and a factor of safety of 5, the equation by substitution becomes

$$\frac{60{,}000 \times 0.3125 \times 87}{27 \times 4} = 151.7$$

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Power and The Engineer

Editorial

Superheated Steam

The question of superheated steam is again brought to the fore by the Bureau of Steam Engineering in a letter addressed to the users, inquiring into their experiences both as to economic results and the kinds of materials required for valves and other fittings. Perhaps a brief review of the history of superheated steam may be of interest. It was in the early sixties that Engineer-in-Chief Isherwood made exhaustive tests on the "Eutaw" to determine the economy of superheated steam. This followed certain preliminary tests on certain steamers plying on the Chesapeake bay and provided with superheaters in the uptakes, where they got the superheating for nothing and in consequence showed large gains in economy.

Now the "Eutaw" had box boilers with vertical water tubes and they showed so good economy and so far reduced the temperature of the furnace gases that there was no use in putting superheaters in the uptake. So a furnace in each boiler was transformed into a superheater, coal being burned for the purpose in that furnace. This at once touched upon one of the fallacies of the advocates of superheated steam, a fallacy that appears to need emphasis every time that the question is brought up. Isherwood found a gain in steam economy of twenty per cent., but when coal was taken as the basis of comparison the gain was reduced to sixteen per cent., which latter was well worth gaining, and superheating became the order of the day. It is worth while noting that the steam pressure was twenty-five pounds and the steam consumption was twenty-six pounds per horsepower-hour. There was trouble with superheating then as there has been since, but the real reason for the failure of superheated steam at that time was the rise of steam pressure as the progress of engineering enabled engineers to build more trustworthy boilers.

The next campaign was in 1877, when Mr. Dixwell published his results, obtained on an experimental engine in the laboratories of the Massachusetts Institute of Technology. Elaborate arrangements allowed the determination of the heat consumption so that the comparison could be based on thermal units, as should always be done in this work. Again the superheated steam showed a large advantage; measured in pounds of steam the gain was twenty-five per cent., but in thermal units the real gain was nineteen per cent. The engine developed only fifteen horsepower and at fifty pounds gage pressure used thirty-five pounds of superheated steam per horsepower per hour. Again the apparent victory was of no avail, for the simple expedient of raising the steam pressure to seventy-five pounds by the gage enabled the engine to show a consumption of only thirty-three pounds of saturated steam per horsepower per hour. Here again it was found necessary to burn fuel to superheat the steam.

At about the same time Hirn was making his celebrated tests on the interaction of the steam and the cylinder walls, and in particular the effect of superheated steam on that interaction. With an engine which developed one hundred and twenty-five to one hundred and fifty horsepower he obtained a steam consumption of about sixteen and one-half pounds of superheated steam and eighteen pounds of saturated steam at ninety pounds gage pressure. Both results are most creditable. Reckoned in steam the gain from superheating amounted to nearly twenty per cent. This again must be discounted to about sixteen per cent. when computed in thermal units. But other tests by Hirn on an engine with steam jackets showed a steam consumption of seventeen and four-tenths pounds of steam, which, if allowance were made for heat in the jacket drains, would come to nearly the figure given for superheated steam. And in 1884, Delafond was able to get an economy of sixteen and five-tenths pounds of saturated steam per horsepower per hour with the aid of steam jackets. So again the superheated steam won in direct test only to be beaten in practice by a more convenient engineering expedient.

Then the matter rested till Schmidt developed his engine for using highly superheated steam and showed such startling results on ten pounds of superheated steam per horsepower per hour in a compound engine of seventy-five horsepower, with one hundred and seventy pounds steam pressure. The conditions of the tests do not allow a ready comparison with the best practice when using saturated steam, so we may turn to the test made by Professor Schröter in 1904 on a compound steam engine using about one hundred and thirty-five pounds steam pressure. He obtained an economy of nine and six-tenths pounds with super-

heated steam and twelve pounds with saturated steam. The apparent gain from superheating was twenty per cent., if the steam consumption is taken as the basis of comparison; but if the heat in thermal units per horsepower per minute be taken, as is proper in this case, the gain is about six per cent. This, by the way, is almost exactly the gain in passing from a compound engine with one hundred and thirty-five pounds pressure to a triple engine with one hundred and seventy-five pounds pressure.

And now, for comparison, let us turn to a test made by Professor Miller on the Chestnut Hill pumping engine, which used eleven and two-tenths pounds of saturated steam per horsepower per minute when developing five hundred and seventy-five horsepower. This engine used two hundred and four thermal units per horsepower per minute with one hundred and seventy-five pounds gage pressure, and the compound engine tested by Schröter used one hundred and ninety-nine thermal units. The difference is of little importance to practical engineers; so it may be said that in the last bout the victory lay with superheated steam, but that a more convenient device robbed the victor of his laurels. Lest the question of the use of superheated steam in triple-expansion engines should appear to be neglected, it may be said that tests show that the gain is too small to be of importance.

That appears to be the condition of superheated steam for reciprocating engines. In every trial there is a decided gain from superheating, usually discounted in a discouraging way when reduced to the basis of coal or heat consumption, but still a very marked gain; but equally good results can be had by some other way which practical engineers prefer.

As for steam turbines, there is a strong claim by the builders that superheated steam shows a valuable advantage, with the mechanical advantage of reducing the amount of water in the lower stages of the turbine. There is reason for treating this claim with respect, even if it be not accepted without reservation. But the conservative engineer has reason for waiting before committing himself on the subject.

The Modern Catalog

All successful business men recognize the value of publicity. Dignified advertising has long since ceased to be experimental—today it is a necessity. While it may take many forms, its best exponent is printer's ink—the press, the circular, the catalog.

Much gray matter—and money as well—are expended in the preparation of catalogs, in the subjects discussed, the manner of presentation, the arrangement, subdivision, indexing, etc. In mechanical execution they are fair to behold. The good taste of the typography, the paper, the color scheme, the engraving and the binding, leave nothing to be desired. Such productions are welcomed and preserved.

They appeal to the manufacturer and the engineer, for in them are to be found many of the latest and best data. Recent advances in the arts and sciences, new apparatus and machines, new uses of existing machines, needed data as to output, efficiencies, costs, the establishment of standards, etc., all these are here ready at hand for use.

It is of vital importance both to the sender and the receiver that such publications should attain their highest usefulness. Unfortunately, there is a reason to believe that these monumental productions often fall short and that there is waste to the sender and annoyance to the receiver. The net result is loss to both.

The modern business man, whether manager, purchasing agent or engineer, must be systematic. He is a man of card indexes and filing systems. He must be able at a moment's notice to put his hand upon any one of many thousand scattered and unrelated bits of data. When he receives a large bound catalog, containing data on boilers, engines, pumps, heaters, valves and pipework, and perhaps also machine tools and contractors' supplies, he is at a loss to know what to do with it. If he files it under any one of its divisions it is lost as to the others. True, he may card index each item in it, as some do, but that is an endless job. Often, perhaps generally, it is placed among other similar general catalogs and is more than likely overlooked when a study is being made of a particular subject.

Two additional objections apply. The space limitations of a general catalog preclude exhaustive treatment of any subject; a concise summary is all that can be expected. Furthermore, all live subjects advance; new ideas, new uses and new patterns tread upon each other's heels. Parts of such a catalog soon become obsolete while other parts are still in full usefulness.

Far better is the modern bulletin system, though it lack somewhat of ponderous dignity. The manufacturer can afford to sacrifice something in color scheme and in early English effect if he thereby gains in clearness and uptodateness. That he also reduces costs and improves efficiencies are incidental, but none the less important advantages.

A separate bulletin for each independent subject is issued whenever the situation warrants. It may be as concise or as diffuse as the occasion justifies. It may be replaced at small expense by a later issue when advances in the art require. Best of all, it may be filed with other literature on the same subject, where it is sure to be found when needed. An inquirer about a special article gets only what he asks for; there is no waste on either side.

Some manufacturers maintain regular mailing lists of persons to whom bulletins are sent as fast as issued. Some of them get data along only those lines concerning which they have filed requests. Cards of acknowledgment are inclosed, the signing and returning of which indicate the recipient's continued interest. Many manufacturers furnish a binder in which all their publications may be preserved together if desired, thus forming a perpetually uptodate catalog.

Good progress has been made toward the standardization of sizes of catalogs. Why not go a step further and standardize their form? The large general catalog, with its multitude of more or less unrelated topics, is a relic of the past. All hail to its successor, the unpretentious bulletin, which serves a single master, but serves him faithfully and well.

Electricity Supply in Great Britain

The economy of aggregation at a point where condensing water is available and fuel accessible without cartage, has resulted in the consolidation of plants of moderate size for the generation of electricity into immense central stations, delivering current at high tension to substations for local distribution. Is it possible that there is to be a swing of the pendulum in the other direction?

The Bow Road station of the Charing Cross, West End and City Electricity Supply Company, at West Ham, London, is equipped with Sulzer steam engines. H. W. Kingston, chief engineer of the company, instead of using his substations merely for transformation, is equipping them with oil engines of the Diesel type, also built by the Sulzers. The units at present installed run from 300 to 450 kilowatts each, but there are some in course of construction having a capacity of 1000 kilowatts. These engines use an oil from which the naphtha and gasolene have been distilled and they produce a kilowatt-hour for about three-eighths of a cent, while at the main station the cost is just under half a cent; both of these figures include all operating charges. When the overhead charges are added, the use of the oil engine substations gives a decided advantage in the reduction of the investment in copper.

All theaters and public places are by law required to have two separate systems of supply. The average price collected is a little above one penny (two cents) per kilowatt-hour.

Gas engines are like some men, they must have either a steady load or a good governor to hold their speed within bounds.

New Power House Equipment

Twinvolute Turbine Pumps

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Turbine pumps have come into popularity because of their efficiency, compactness and adaptability to all methods of drive.

The Watson-Stillman Company, New York City, holds the exclusive rights to manufacture the Twinvolute type of turbine pump. The difference between this type and the ordinary concentric turbine pump, in which the waterway is concentric with the runner, is shown in Fig. 1. In the Twinvolute the waterway at the edge of the casing varies in size from its fullest width at the outlet to the narrowest point at the opposite side.

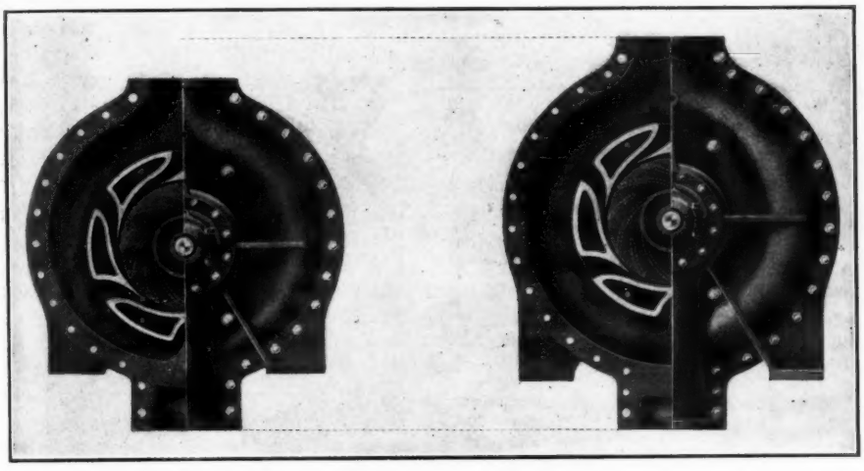


FIG. 1. RELATIVE SIZES OF TWINVOLUTE AND ORDINARY CONCENTRIC PUMPS

With this design many advantages are secured. It is assumed by the makers of turbine pumps of the concentric-waterway type that when the water leaves the diffusion-vane outlets the water divides and passes in equal amounts around and through the concentric ways. Theoretically this is right, but it has been proved by experiment that the parting is seldom at the mid-point; that is, diametrically opposite to the pump-discharge outlet. In fact, no two pumps have the dividing point in the same place. This is due, probably, to inaccuracies and variations in the smoothness of the inside surface of the pump shell.

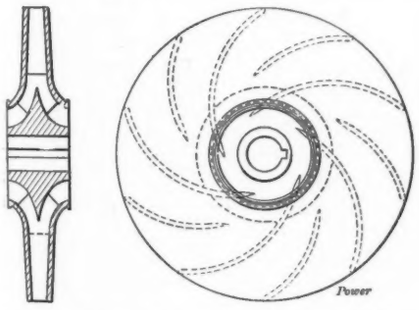


FIG. 2. IMPELLER OF SINGLE-STAGE TWINVOLUTE TURBINE PUMP

If the actual dividing point does not correspond with the theoretical, it is obvious that the water must do more work on one side than on the other in reaching the discharge outlet. This results in a drop in the pump efficiency.

The water leaves the discharge outlet of any turbine pump at a predetermined velocity, generally 10 to 12 feet per second. If, when the water passes through the diffusion-vane outlets, it enters a waterway concentric in shape and of an area equal everywhere to that at the discharge outlet, there is a constant increase and decrease in velocity and pressure in the waterway.

The Twinvolute waterway avoids these errors of unequal work, pressure, velocity and objectionable eddies, and adds good features not obtainable with the ordinary construction. As the water leaves the diffusion vanes it is compelled by the construction of the volutes to part at the correct center. Each volute must carry an equal amount of water; hence, the distribution of work is uniform, and every drop of water is compelled to reach the pump discharge by the shortest course. The Twinvolute is so graduated

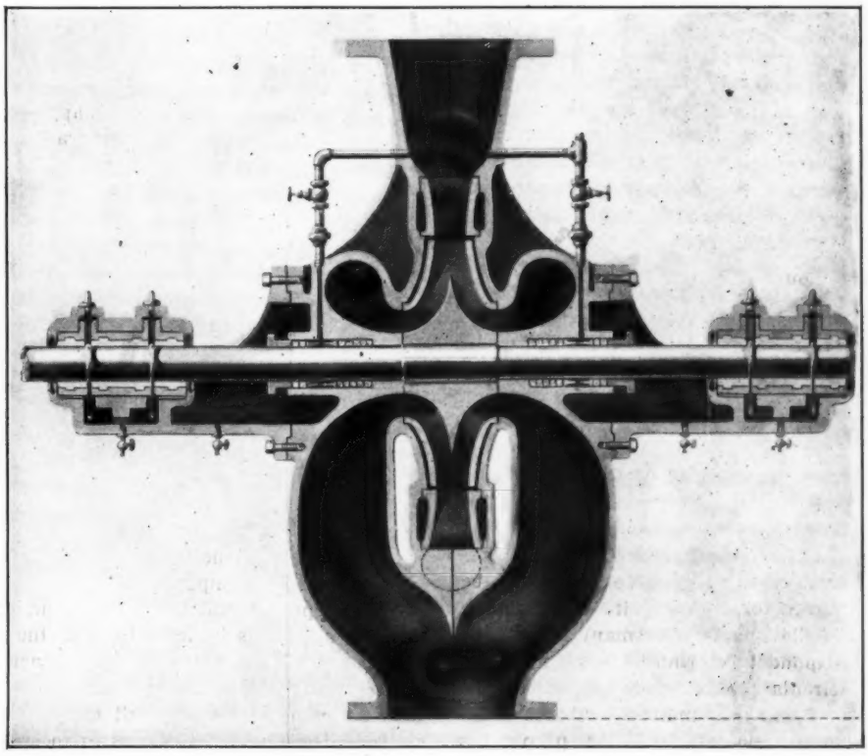


FIG. 3. CROSS-SECTION OF SINGLE-STAGE PUMP

in sectional area that as the water issues from the diffusion-vane outlets, the additional amounts of water on their way to the discharge outlet are given ample space and variation of pressure at the different diffusion-vane outlets is prevented. The predetermined velocity is, therefore, constant; there are no whirls or eddies and the efficiency must remain at the maximum. Also, with this construction, the size and weight of the pump are materially decreased, thus lowering the first cost. A still further saving is effected by the reduction in the size of the foundations and the amount of floor space required. The internal area or wetted surface is less than with the concentric waterway. The path of travel of a particle of water from the eye of the impeller out through the diffusion vanes and twinvolute to the discharge opening is sufficiently less than in other forms of construction to make a noticeable reduction in the friction loss. Further, because of the twinvolute, the center of gravity is lowered and the vibration reduced. This means longer life for the pump and an absence of noise, which with some types may be troublesome.

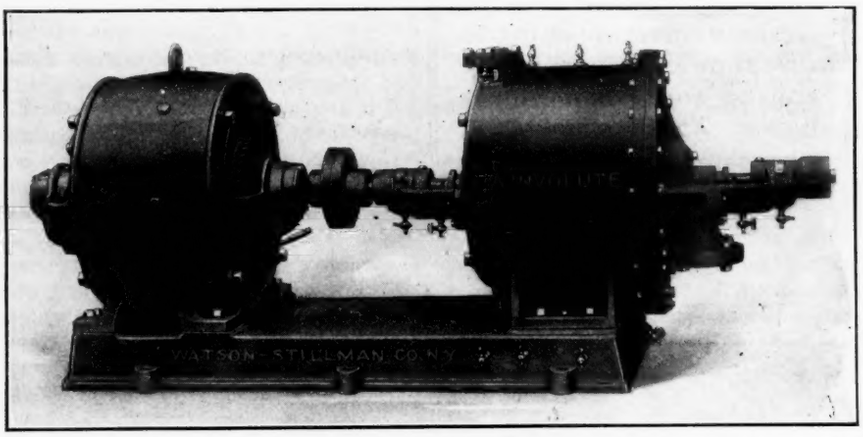


FIG. 4. MOTOR-DRIVEN TWINVOLUTE PUMP FOR BOILER-FEED SERVICE

As shown in the cross-section, Fig. 3, water in the suction pipe must divide in the Y passage of the single-stage pump and flow in equal amounts into each side of the impeller.

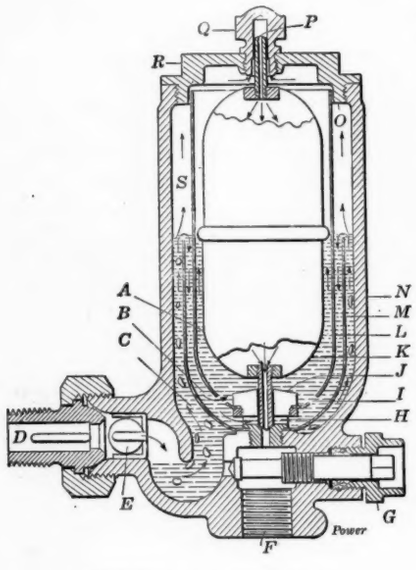
Symmetry is embodied in the impeller design so that as the water enters the eyes on each side, there is equal pressure in opposite directions. The pressure being equalized there is a natural balance and end thrust is avoided.

The divided suction is especially desirable in high-speed and large-size pumps for, by reason of the double openings, the impeller may be made smaller in diameter than in pumps where all of the water must flow into one set of openings. This means a smaller pump and a consumption of less power.

The Watson-Stillman Company manufactures turbine pumps for capacities of from 2 to 150,000 gallons per minute, for suction lifts of from 1 to 25 feet, for heads of from one pound up to that equivalent to a discharge pressure of 500 pounds and for speeds of from 50 to 3000 revolutions per minute. The pumps may be operated by gearing, belts, silent chains, or direct-connected electric motors, gas engines, steam engines, steam turbines or water turbines. The pumps may be used for practically any service, such as heating systems, water works, hydraulic sluicing, condensing systems, storage tanks, boiler feed, fire protection, irrigation and mine drainage.

Monash Radifier Valve

Two great faults that heretofore have been encountered in the erection of vacuum steam-heating systems are claimed to be done away with by the installation of the Monash radifier; that is, the noise caused by the chattering of the float and the removal of the valve for cleaning.



MONASH RADIFIER VALVE

The still body of water *A*, see illustration, in which the float is incased, prevents the chattering of the float, due to air or water currents. The seat *J* can be unscrewed and the interior of the valve taken out and thoroughly cleaned without disconnecting the valve from whatever it is attached to.

The valve is attached to the return end of the radiator or heating device by the union nipple and to the return pipe at the outlet *F*, which is connected to the vacuum pump.

The air and water of condensation are drawn by the vacuum from the radiator into the radifier through the inlet *D*, past the ball check *E*, down into the gooseneck trap, where any heavy dirt or foreign matter is deposited, which may be blown or drawn through the bypass valve into the return.

The air and water of condensation pass upward in the chamber *C*, when the air separates from the water and is drawn through the chamber *S* and the opening *O* to the hollow-float guide pin *P*, through the float *K* down to the return through the outlet *F*.

The water of condensation, being drawn up into the body of the valve, rises to the upper edge of the outer shell *M*, flooding the chamber *B* and also gradually filling the chamber *A* by slowly passing through the openings *I*. This opening is of small dimension and prevents the sudden rush of water into the chamber *A*, thus breaking the current or force of the water, and prevents the chattering of the float.

The water gradually rises around the float to a point near the seam, when it raises the float *K*, thereby opening the discharge passage through the valve seat *J*.

While the valve is open the chamber *B* is drained by the water passing through the opening *H* in the valve seat and to the return pipe through the outlet *F*, and since the discharge is over the top of the shell *M*, the oil, grease or floating matter is skimmed and is carried off with the water.

The receding water in chamber *A* keeps the float buoyant until the valve is closed, thereby insuring a noiseless operation of the float.

The phosphor-bronze ball check *E* is held in place by a copper basket and, seating in the union nipple, prevents the vacuum in the radiator reacting on the vacuum in the return when the radiator supply valve is closed.

To clean the passageway of the float, the cap *Q* may be unscrewed and a wire forced through the openings in the float guide stem.

The bypass valve is protected by a locked shield feature *G* which prevents tampering with the valve.

This valve is made by the Monash-Younker Company, 1407-17 W. Jackson boulevard, Chicago, Ill.

The Perfection Calorimeter

The testing of coal, coke and liquid fuels to determine their heat content can be accomplished by anyone of ordinary intelligence and with very little expense by the aid of the instrument described and illustrated herewith.

In Fig. 1 is shown a sectional view of the Perfection calorimeter, manufactured by the Western Meter Company, Davenport, Ia. The metal can *A* is filled with water; *B* and *C* are asbestos cylinders for insulating the instrument from the effects of atmospheric changes while the determination is in progress.

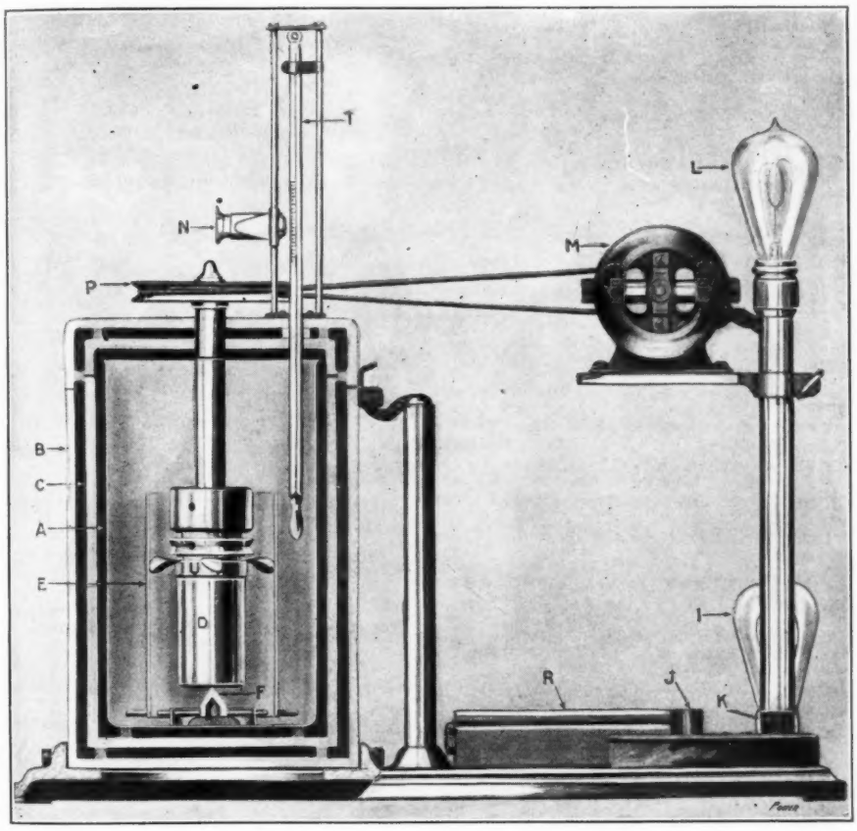


FIG. 1. THE PERFECTION CALORIMETER

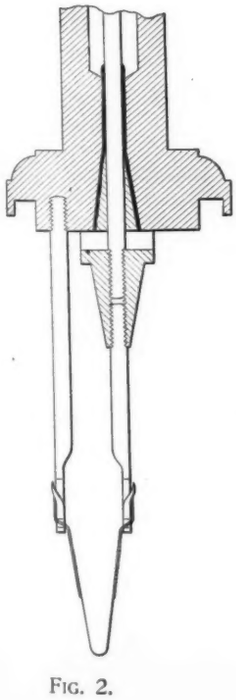


FIG. 2.

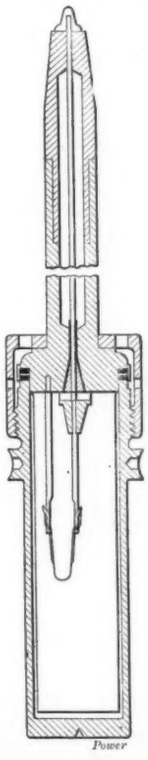


FIG. 3.

The bomb *D* is equipped with an improved combustion chamber in which is placed a weighed quantity of coal or fuel to be tested. The deflector *E* deflects the currents of water, thereby facilitating the equalization of the temperature. The bomb rotates on the pivot *F*. A light which indicates when the charge is ignited is shown at *I*; *J* and *K* are switches for controlling the motor and ignition, and *L* is a light in series with the motor. A telescopic lens *N* is provided for reading the thermometer *T*.

The auxiliary chamber in which the combustion takes place is entirely surrounded by air, so that the sides of the chamber become red hot and cause even the most refractory particles of fuel to be consumed. The instrument is especially adapted for the testing of liquid fuels.

Fig. 2 shows the method of igniting the charge and of fastening the fuse wire *G*. At each end the fuse wire is wound into loops and is attached to hook-shaped lugs. Fig. 3 shows the fuse wire as it appears in the bomb.

When ready to test, weigh out exactly one-half gram of commercially dry coal and place it in an oven to dry. When the coal has been dried it is transferred to the combustion chamber which has been removed from the bomb *D*, Fig. 1. To the coal one measure of chemical is added before the cylinder is replaced in the bomb, the cap adjusted and the whole vigorously shaken. The can *A* is removed from the instrument and filled with exactly 2000 cubic centimeters of water. It is then replaced, the bomb *D* inserted, the cover on the instrument put in place and the thermometer *T* inserted so that the mercury bulb is located at about the center of the can *A*.

The pulley *P* is connected by a small belt to the motor, which rotates the bomb *D* and stirs the water.

The temperature of the water will equalize in about three minutes, it is claimed; the fuel charge can then be ignited by pressing the button *J*, when the heat of the combustion is imparted to the water, and the rise is recorded by the finely graduated thermometer *T*. The rise is then multiplied by a simple constant, and the result is the exact number of British thermal units per pound of fuel tested.

The instrument is complete, mounted on a base 11x22 inches, and is ready for use when connected to any lamp socket available.

The Mechanical Engineers at London

The mechanical engineers left Birmingham on Thursday morning and after visits to Warwick, Stratford-on-Avon and Kenilworth, were conveyed by special train to London, where they arrived in time to be received by President Aspinall of the British Institution at a conversazione at the Institution home. The feature of the evening was a lecture by Doctor Hele-Shaw on "Stream Lines as Affecting Aviation." Stream lines were projected upon the screen by injecting coloring matter into a sheet of glycerin flowing between two glass plates in the lantern, and the area of reduced pressure behind a plane in the stream was very effectively shown by using a less viscous fluid.

On Friday morning a session for the consideration of professional papers was held at the Institution of Civil Engineers, and in the afternoon teas were given by Sir John I. Thornycroft and William Maw. In the evening President Aspinall entertained at dinner, some five hundred covers being laid. The usual line of toasts were drunk, that to the American Society of Mechanical Engineers being proposed by Sir William White and ably responded to by Dr. F. R. Hutton.

On Saturday the party was taken to Windsor and for a sail upon the Thames between Windsor and Marlowe, reaching London by special train in the early evening. A visit to the Japanese exposition on Saturday evening completed a day

every minute of which was full of interest.

On Sunday evening a special service was held in Westminister Abbey and an opportunity afforded to see the memorial window to Sir Benjamin Baker, who was an honorary member of the American Socity.

It was an exhausted party which dispersed at the end of the program, which, for the Americans, had included two weeks of incessant entertainment and activity. Everybody was delighted with his experiences, and full of appreciation of the generous and spontaneous hospitality which made the visit such a mélange of pleasurable impressions.

BOOK RECEIVED

"ELECTRIC WIRING." By Joseph G. Branch. Branch Publishing Company, Chicago, Ill. Cloth, 287 pages, 5⅛x 7¾, 93 illustrations, indexed. Price, $2.

PERSONAL

P. M. Chamberlain and S. S. Howell, both of whom were formerly connected with the Under-Feed Stoker Company of America, have formed a partnership and are engaged in general consulting-engineering work in Chicago. Their offices are in the Marquette building.

A. F. Norcross has succeeded Hubert E. Collins as general manager of the Engineering Supervision Company, 45 West Thirty-fourth street, New York City.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

REVOLVING GAS ENGINE. George Haniquett, Longbeach, Cal. 965,385.

WINDMILL. Henry I. Hawxhurst, Los Angeles, Cal., assignor to Hawxhurst Wind Turbine Power Company, Los Angeles, Cal., a Corporation of California. 965,387.

STEAM TURBINE. Rudolph Schlatter, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 965,412.

COMPOUND STEAM PUMPING ENGINE. Alexander England, Wilkinsburg, Penn., assignor to the Westinghouse Air Brake Company, Pittsburg, Penn., a Corporation of Pennsylvania. 965,573.

ROTARY ENGINE. Richard Lundquist, Laguna de Terminos, Mexico. 965,640.

EXPLOSIVE ENGINE. Jay F. Woolf, Minneapolis, Minn., assignor of one-half to Ellis J. Woolf, Minneapolis, Minn. 965,975.

COMBINED LIQUID AND GAS TURBINE. Carl E. Brockhausen, Chicago, Ill. 964,985.

HOT-AIR MOTOR. Guillaume Mann, Paris, France. 966,032.

CURRENT MOTOR. Edgar A. McClung, Dayton, Wash. 966,041.

ROTARY CURRENT MOTOR. Edgar A. McClung, Dayton, Wash. 966,042.

INTERNAL COMBUSTION ENGINE. Alphonse Butsch, St. Lucia, British West Indies. 966,309.

GAS TURBINE. Leo Samoje, Antwerp, Belgium. 966,363.

INTERNAL COMBUSTION ENGINE. Erik Anton Rundlof, Stocksund, near Stockholm, Sweden. 966,362.

BOILERS, FURNACES AND GAS PRODUCERS

GRATE. Fred Hilburger, Castle Rock, Colo. 965,186.

GARBAGE BURNER AND WATER HEATER. William G. Ross, Chicago, Ill., assignor of one-half to Frank Sutherland, Chicago, Ill. 965,229.

CRUDE-OIL BURNER. Oskar Kirschen, Kolomea, Austria-Hungary. 965,305.

GRATE. John Levey, Chicago, Ill., assignor to John Williamson, Chicago, Ill. 965,589.

OIL BURNER. Arthur H. Light, Los Angeles, Cal. 965,719.

OIL BURNER. William E. Weathers, Frederick, Okla. 965,775.

DIRECT-ACTING PUMPING MACHINERY. John Hutchings, London, England. 965,819.

DISTRIBUTING VALVE. John Hutchings, London, England. 965,820.

CRUDE-OIL BURNER. John A. McCoach, La Cygnes, Kan. 965,831.

STEAM BOILER. Thomas E. Durban, Erie, Penn. 965,883.

BURNER FOR FUEL OILS. Albert W. Thompson, Manchester, N. H., assignor of one-fourth to Harry E. Blanchard, Manchester, N. H. 965,963.

SMOKE-CONSUMING APPARATUS. William Ramsay Marshall, Oldham, and Thomas Hargreaves, Ashton-under-Lyne, England, assignors to Victory Smoke Consumer and Fuel Economizer Company, Ltd., Oldham, England. 966,109.

SPRAY BURNER. Archie T. Rigg, Altoona, Penn. 966,124.

STEAM GENERATOR. William George Hay, Tuebrook, Liverpool, England. 966,201.

FURNACE. John Harrigan, Brooklyn, N. Y. 966,200.

FURNACE. Carl Nordell, Stamford, Conn. 966,233.

POWER PLANT AUXILIARIES AND APPLIANCES

OIL CAN. Luther M. Graves, Philadelphia, Penn., and James Woodward Allen, Lumberton, N. J. 965,578.

HOSE COUPLING. Joseph W. Ferguson, Quincy, Mass. 965,286.

THROTTLE. Jonathan Peterson, Brooklyn, N. Y. 965,322.

PISTON AND PISTON VALVE. Robert Allen, Caversham, England. 965,355.

SPARK PLUG. Robert T. Estes, and John P. Inman, Midland, Tex. 965,380.

TURBINE PACKING. Robert A. McKee, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 965,399.

COMBINED THROTTLE VALVE AND OILER. Lewis C. Bayles, Johannesburg, Transvaal, assignor to Ingersoll-Rand Company, New York, N. Y., a Corporation of New Jersey. 965,554.

IGNITION TIMER FOR INTERNAL COMBUSTION ENGINES. Francis C. Mason, Grand Rapids, Mich. 966,034.

VALVE. Eugene J. McCarty, Clinton, Mass. 966,040.

VALVE MECHANISM FOR INTERNAL COMBUSTION ENGINES. Hermann Steingassner, New York, N. Y., assignor, by direct and mesne assignments, to Alfred Hutter, New York, N. Y. 966,367.

VALVE. James P. Haig and Alvin H. Porter, Grand Rapids, Mich. 966,198.

STEAM-POWER APPARATUS. Wilbur C. Anderson, Canton, Ohio. 965,850.

CARBURETER. Paul Bustard, Frederick, Okla., assignor to Eskew H. Archer, Frederick, Okla. 965,867.

SPARKING PLUG FOR EXPLOSIVE ENGINES. Cyrus F. Jones, Oregon, Ill. 965,585.

METALLIC PACKING. William B. Claflin, New York, N. Y. 965,688.

PRESSURE GOVERNOR FOR ELECTRICALLY OPERATED PUMPS. Francis L. Clark, Pittsburg, Penn., assignor to Standard Traction Brake Company, a Corporation of New Jersey. 965,689.

DRAFT REGULATING DEVICE FOR FURNACES. Wilber P. Hall, Pembroke, N. Y.: Alida L. Hall, administratrix of said Wilber P. Hall, deceased. 965,815.

ELECTRICAL INVENTIONS AND APPLICATIONS

COOLING TOWER. Donald Barns Morison, Hartlepool, England. 965,116.

VALVE. Charles Walker, Knoxville, Tenn. 965,129.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., V. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.